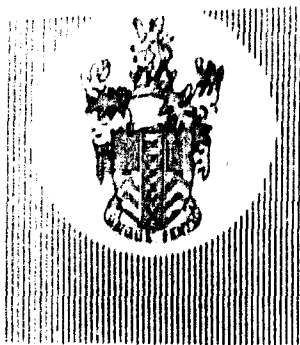


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Report No. 87018

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**ROYAL SIGNALS AND RADAR ESTABLISHMENT,
MALVERN**

**ASSESSMENT OF THE
PHILIPS USFA BV 7 mm COOLING ENGINE**

Authors: D P Laycock and W J A Powell

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RSRE

Malvern, Worcestershire.

May 1988

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ROYAL SIGNALS AND RADAR ESTABLISHMENT

REPORT 87018

TITLE: ASSESSMENT OF THE PHILIPS USFA B.V. 7MM COOLING ENGINE

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SUMMARY

Philips Usfa B.V. Stirling cycle split-mode cooling engines, sampled from development, pre-production and production batches, have been extensively tested at RSRE. Measurement of cryogenic performance at normal and high ambient temperatures in run-stop-run and continual-run regimes have been made. Representative detector encapsulations have been interfaced to provide typical combination heat loads. One pre-production unit has been installed and operated successfully in a Thermal Imaging Common Module (TICM) Class 2 Scanner. Detector microphony-noise measurements were made and compared with those recorded when the same array was cooled with high pressure pure gas liquified by a Hymatic self-regulating mini-cooler. Initial problems of contamination of the helium gas and of the thermal interface with the detector encapsulation have been highlighted and addressed by the manufacturers. A production standard unit has achieved over 5,000 hours intermittent and continuous running during a period of 12 months.

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1. INTRODUCTION

The Philips Usfa B.V. UA 7039 to UA 7044 series of Stirling cycle cooling engines (termed by them "Split Mini-coolers") are a third generation derivative of their field-tested military Mono-block UA 7011. The UA 7041 has a 6.8mm diameter cold finger with a nominal cooling capacity of 500mW at 80K when operated in an ambient temperature of 70°C and is capable of interfacing with Mullard dual-purpose (engine or Joule Thomson cooled) encapsulations. RSRE Malvern has a long-standing MOD commitment to test manufacturers' coolers for Project Office.

The engines obtained from Philips Usfa B.V. for assessment have been samples of their build-standards in development, pre-production and production stages. In a development programme, early samples are used to establish the technology which production units must maintain at a proven standard and for an economic price. A test programme follows this progression, moving from the initial acquisition of test experience to the practised assessment of the uniformity of performance and recognition of any enduring design/build limitation. The speed at which shortcomings are identified, corrected and improved components supplied is apparent in the test history overall. Since the early development models were steadily superseded by pre-production units, this report gives the later assessments which were most significant in establishing performance and identifying areas for improvements to be carried out in the production standard.

2. GENERAL DESCRIPTION

Philips "Split Mini-coolers" currently comprise a compressor module connected via a small-bore pipe to cold fingers ranging in diameter from 4.75 to 9.90mm. These cater for cooling heat loads of 250mW to 1000mW at 80K. They operate on the Stirling cycle, a thermodynamic principle described in many papers e.g. References 1, 2 and 3.

2.1 Cooling Engine

The following description is based on accounts published in the open literature. The engine, Figure 1, consists of a compressor module, a cold finger designed to mount into the detector encapsulation and an interconnecting pipe 2.4mm diameter by 300mm long which may be conformed to suit the installation layout. These three components are welded together and can be dismantled by grinding away the joins then repaired and re-welded.

2.1.1 The compressor contains a piston which is directly coupled to the moving coil of a permanent magnet linear motor. The samarium-cobalt magnet maintains a uniform radial magnetic field in a cylindrical gap, and the coil operates in this gap, driving the piston in response to an AC current. The piston/coil assembly is mounted on a spring which determines the mid-stroke position. All forces on the piston are axial ones: no side-loads are imposed on the piston seal, thus minimising wear and hence contamination of the gas. The compressor unit is laser-welded to minimise helium out-diffusion. Compressor vibration is reduced by the incorporation of a dynamic vibration absorber in the form of a mass suspended between opposing coil-springs.

2.1.2 The pipe connecting the compressor to the cold finger increases the design flexibility of the system, as well as serving to decouple the compressor vibration from the encapsulation.

2.1.3 The moving displacer in the cold finger contains a gauze regenerator, and is mounted on a coil spring at the warm end. The displacer is driven at the piston frequency by the pneumatic force resulting from the pressure drop across the flow-resistance of the regenerator packing. In order to obtain cooling, the phase lead between the displacer and the piston is achieved by tuning the displacer-mass/spring system to a higher resonant frequency than that of the operating frequency of the engine. Thus the displacer is at the bottom of its stroke when a relatively large mass fraction of the gas is at the end of the cold finger during the expansion phase resulting in abstraction of heat, that is, cooling. The heat is rejected at room temperature during the compression phase with the displacer at the top of its stroke. The regenerator mesh in the displacer, by alternately absorbing and rejecting heat from the working gas, establishes the temperature differential between the cold and warm ends of the cold finger. The forces on the displacer, like those of the compression piston, are axial, thus the seals suffer little wear.

2.2 Detector Interface

The cold finger has a flange for mounting it to the detector encapsulation. A recess is provided for locating the encapsulation concentrically and a groove contains a silicone-rubber O-ring to prevent the ingress of atmospheric moisture. A spring-loaded pad, mounted on the end of the cold finger, presses against the underside of the detector substrate platform to achieve thermal transfer. A smear of silver powder loaded silicone grease supplied with the cooling engine aids intimate contact.

2.3 Heat Dissipation

Philips recommend that the surface temperatures of the compressor module and the cold finger should not exceed the ambient temperature by more than 10 degrees. The operating ambient temperature range expected is -40°C to $+70^{\circ}\text{C}$. Adequate heat sinking must be provided by the user since none is present on an engine as supplied. The method used in our tests was to fit aluminium-alloy black anodised finned cylinders clamped to the compressor casing and to the warm end of the cold finger in good thermal contact. Figure 2 shows the engine component parts and disassembled finned-jackets. Figures 3(a) and (b) are outline drawings of the complete assemblies of the integral balancer and latest demountable balancer models respectively. A small cooling fan displacing some $50\text{m}^3/\text{hr}$ of ambient air was initially used to aid heat dissipation from the engine.

3. TEST EQUIPMENT AND SETTING-UP

3.1 Test Encapsulation

Laboratory assessment of cooling engine performance was formerly made by fitting a vacuum test-dome and monitoring the condition at the end of the cold finger with suitable instrumentation. This technique has now been superseded by testing with a production-built detector encapsulation with bias loads applied via the detector array in order to provide representative data to potential users. Mullard R185 encapsulations provided realistic heat loads with their Sprite detectors biased to give a maximum of 500mW.

3.1.1 The heat load of the encapsulation used for the tests was determined at the onset and periodically checked. Measurements were made at Mullard Ltd, Southampton by using the latent heat of boil-off method with liquid nitrogen and liquid oxygen over a range of ambient temperatures. Figure 4 shows the spread of the measured data and 'best-fit' lines. A typical value, at 20°C, for the heat load of an R185 encapsulation is 100mW at liquid nitrogen temperature.

3.1.2 Temperature sensing diodes (TSD's) are provided on the substrate in IR detector encapsulations for built-in-test-equipment (BITE) monitoring of array operating temperature and thus cooler efficiency. In engine-compatible encapsulations, the diode also provides a means to control the compressor in order to achieve stable cooling performance, typically plus or minus 1 degree K. The TSD's, BC557's in this case, were calibrated by measuring the forward voltage with a 1mA current source at the liquid nitrogen and liquid oxygen fixed points and at room temperature. These measurements, with allowances for a slight temperature difference through the substrate (1.6 degrees K), were within 1mV of a test performed at Mullard Ltd, Southampton when a variable cryostat was used. Figure 5 shows the TSD calibration.

3.1.3 The only regions of the encapsulation and cold finger which should contact each other when the components are mated are the mounting flanges and the cold tip pad sprung-loaded against the substrate platform. This requires the cold finger and flange to be closely concentric and orthogonal: similarly for the bore and flange of the encapsulation. These parameters were checked to ensure that they were within their manufacturer's tolerance. The encapsulation bore was 7.24mm diameter with a departure from rectilinearity resulting in 0.10mm offset at the substrate - on the top limit of the tolerance zone. The critical region on the cold finger is the copper cap to which the thermal coupling is attached. Its diameter was out of round and measured 6.840mm minimum, 6.900mm maximum for the major length of the cap and, not being parallel, measured a maximum of 6.940mm over a small area. The tolerance limits on the drawing gave 6.80mm minimum, 6.90mm maximum. The offset at the cold end with respect to the flange recess was 0.03mm. The drawing gave a total perpendicular tolerance zone of 0.10mm, that is, 0.05mm permissible offset from the vertical.

3.1.4 The interface force required to compress the spring thermal coupling on the cold finger against the blind-end of the encapsulation bore was established as 8.9 Newtons. Mullard Ltd recommend a maximum interface force of 22.2 Newtons for the R185 encapsulation. A thin smear of silver powder loaded silicone grease supplied with the cooling engine was applied to the pad on the thermal coupling in order to improve contact with the high thermal conductivity substrate platform. The sealing-ring for the cold finger flange was given a smear of silicone high vacuum grease, the encapsulation carefully located over the finger and held in place with three equi-spaced clamps.

3.2 Heat Dissipation

Cooling fin assemblies, manufactured from aluminium alloy and black anodised, were clamped on the compressor module and on the warm end of the cold finger. Silicone heat sink compound grease was used to improve thermal conduction between the mating surfaces. Secondary heat sinking was obtained by mounting the complete assembly on an aluminium base plate with the compressor and cold finger axes in line. The majority of the tests were performed with the axes in the horizontal plane, Figure 6. Airflow over the fins was initially provided by a 3W Boxer fan.

3.3 Instrumentation

The laboratory set-up used for the assessment is shown schematically in Figure 7: a Yew hybrid analogue/digital recorder, (not shown), Feedback Wattmeters, Keightly DVM and Thurlby DVM, were used to monitor and record input power to the engine, bias power to the array, TSD voltage, cooling engine surface temperature and ambient temperature. The compressor required a 50Hz AC input which was provided from the 240V mains supply via a 6:1 step-down transformer and Variac control. The elements of the Sprite detector were connected to a variable DC supply to provide bias loads. A constant current source was used to drive the TSD on the array substrate, and the forward voltage monitored on a 5.5 digit DVM. The temperatures at the surface of the compressor and the warm end of the cold finger were monitored by thermocouples.

3.4 Environmental Oven

For assessment at elevated ambient temperatures, the complete cooling engine assembly was placed in a large oven controllable to plus or minus 1 degree K. A fan was positioned to maintain forced airflow (at oven-ambient temperature) through the cooling fins.

4. TEST PROCEDURES

The design requirement cooling capacity of the 7mm UA 7041 "Split Mini-cooler" is nominally 0.5W. This is the sum of the encapsulation heat load and the bias load (for photoconductive detectors in particular) applied to the array. Thus at nominal ambient, the engine must cope with up to 400mW array bias load, and at +70°C circa 300mW applied bias. Since all cooling engines can be expected to lose performance gradually as helium out-diffuses and wear occurs, and evacuated encapsulations becomes "softer" with age, the assessment measured performance at array bias loads greater than those specified by Philips. These data were aimed at giving confidence to users that there was performance to spare, especially under standard ambient conditions. Data given are for a production-standard unit. The results presented show the input requirement to the cooling engine as a function of Watts. The relationship of input volts AC 50Hz and current with respect to input power at ambient temperature of 20°C and 70°C is shown in Figure 8.

4.1 Performance at 20°C ambient

4.1.1 Cooldown times were measured by maintaining a constant input power to the compressor and recording the TSD voltage, hence substrate temperature at 5 second intervals. The array was not biased. Figure 9 shows the variation in cooldown time for the range of input powers 20, 30, 40 and 50W.

4.1.2 The input power and voltage requirement to maintain a stable cryogenic temperature of 80K at the substrate was measured for array bias loads of 0, 50, 100, 150, 200, 250, 300, 350, 400, 450 and 500mW. The measurements were repeated at 90 and 100K. Figure 10 shows the bias power (mW) plotted against input power (Watts) for the related cryogenic temperature. It is emphasised that the encapsulation heat load is not included in these figures.

4.2 Performance at higher ambient temperature

4.2.1 The cooldown time measurements, 4.1.1, were repeated at 30°C, 50°C, 70°C: Figures 11, 12 and 13 show typically cooling engine performances of the UA 7041.

4.2.2 Tests detailed in 4.1.2 were repeated at 30°C, 50°C and 70°C. Figures 14, 15 and 16 show the performance. Bias loads in excess of 300mW at 70°C push the engine progressively beyond its cooling capacity at 80K, since the encapsulation load and efficiency of thermal transfer account for an additional circa 200mW effective heat load. The cooling engine combination catered for 700mW, but the lowest temperature reached was 95K. An additional 12W fan was needed to keep the cold finger warm end temperature to within 10 degrees of the imposed ambient under these conditions.

4.3 Detector Noise-Voltage Measurements

Because the compressor operated relatively close to the encapsulation, any electromagnetic interference with array noise-voltage was established and compared with that observed when Joule Thomson minicooling was used. The R185 is a cooling engine/Joule Thomson compatible encapsulation.

4.3.1 Detector noise measurements were made in a screened room. One of the elements of the Sprite array was connected to a TICM Class II preamplifier, and the output fed to a Hewlett-Packard Spectrum Analyser which performed the spectral plot, normalising the output to nanovolts per root Hz ($\text{nVHz}^{-1/2}$). The analyser was set-up to avoid 50Hz and its harmonics in order to avoid mains pickup imposing spikes on the noise plot. The plots of the element, made with 4mA bias current applied, when cooled to 80 and 100K by the UA 7041 cooling engine and by a Hymatic MAC 217 self-regulating cooler are shown in Figure 17.

5. TEST PROGRAMME

The tests outlined in Section 4 are applied to any cooling engine presented for assessment: they establish the key requirements of cryogenic performance. When this baseline is satisfactory, reliability and durability can be assessed while the unit undergoes run-stop-run and continual-run regimes. These do not represent the full duty-cycle (which is specific to a system in a climatic zone) but serve to stress the engine realistically and identify any significant problem areas such as mechanical wear, helium loss and degassing contamination within the sealed unit.

Following the initial tests on cooldown time and stability of array temperature (which equate to acceptance tests), the engine undergoes run-stop-run sequence totalling 16 hours running time per day, Figure 18. These are interspersed with random length runs and at least one extended period of operation. The bias load applied during the majority of these tests is 100mW - a nominal value for a UK TICM Class II Sprite detector of 80mW plus 25%.

5.1 Development Models (Serial Nos. 053, 054 and 068)

5.1.1 The first spit-mode cooling engine obtained from Philips Usfa were serial numbers 053 and 054, both extensively measured but only with instrumentation caps. Number 068 was interfaced with several detectors and underwent a series of tests, summarised in the "motorway map", Figure 19. It completed a total running time of 2330 hours including one continual period of 1000 hours.

5.1.2 After 270 hours, the performance had deteriorated and required an increase in input power to maintain the performance at 80-100K. Investigation of the encapsulation/cold finger assembly gave no obvious clue at first, but careful re-assembly restored performance. The cause was identified as the soft copper flange on the cold finger which was distorted. The occasional small

adjustment to the clamping screws which checked tightness had probably resulted in the cold finger either touching the side of the bore and being thermally shunted; or distorting the finger and interfering with the free movement of the displacer.

5.1.3 At this period, too, it became evident that the cooling engine showed a tendency to stick at various cold-end temperatures during the cooldown run, usually between 190-210K. Several changes of encapsulations and additional checking with an instrumentation cap failed to reveal the reason. The engine would unpredictably regain its full performance and run for long unsupervised periods.

5.1.4 The engine was interfaced at one stage with a UK TICM Class II scanner and successfully operated (see Appendix 1). Its return to Philips after some 2330 hours of operation was not for any specific failing, but to make way for tests on a pre-production unit. The possibility that gas contamination by a species with solidifying temperature around 200K had been noted and residual gas analysis at Eindhoven was suggested.

5.2 Pre-production Units (Serial Nos. 0108 and 0370)

5.2.1 A total of 5033 running hours was accumulated by the cooling engine, summarised in the 'motorway map', Figure 20. After 100 hours, the performance showed a deterioration similar to 068: a fall-off in cooling capacity and intermittent reluctance to achieve better than 190-210K.

5.2.2 Consider firstly the fall-off in performance: the flange on the cold finger had been changed from copper to stainless steel, thus eliminating distortion, therefore contact between bore and cold finger was suspected. Measurements of these items on acceptance had shown that the copper cap on the cold finger was 6.97mm diameter, at that time within the manufacturer's tolerance, within a bore of 7.23mm diameter. Thus a radial clearance of 0.13mm was possible, provided that the finger was orthogonal to the flange and concentric. The copper cap was covered with a thin smear of engineer's marking blue. Assembly/disassembly resulting in a witness mark on the cap and blue left within the base at the upper end. Very careful reduction of the diameter of the cap by filing it effected a clearance, but the practice is not recommended.

5.2.3 The intermittent tendency to stick at circa 200K was overcome by carrying out an extended cooldown procedure, gradually increasing input power as the substrate temperature fell. By this means, cooling capacity was achieved and the bulk of the tests carried out. After 3260 aggregated hours, the cooler was left to run indefinitely and during the ensuing 1773 hours its performance fell off until it needed 39W input rather than 22.5W. The compressor finally became noisy in the balancer region and it was suspected that the mass/spring had worked loose. This engine, too, was returned to Philips Usfa for destructive analysis.

5.2.4 Cooling engine Serial No. 0370 was not tested. Its build history was similar to 0108 and the evidence of contamination had been confirmed by residual gas analysis as water-vapour outgassing within the compressor and finally freezing out at various positions within the cold finger which has a thermal gradient from cryogenic tip to warm end. Ice-crystal formation occurred at positions along the regenerator which depended upon the input power, ambient, fan cooling position etc. This explained the variations in sticking temperature and the ability to overcome it by slow cooldown. The outgassing/de-sorbing component within the compressor was identified and corrective action taken for the subsequent production build.

5.3 Production Units (Serial Nos. 0414 and 1108)

5.3.1 The graphs of performance Figures 9 to 16, are typical of the data from 0414; it gave no problems in fast cooldown and achieved its specified cooling capacity. Overtesting in the sense that it has been run for periods with 500mW bias applied thus providing a total heat load of 650mW, did not appear to have degraded it significantly. However, after some 500 hours of running time, it became apparent that 20% increase in input power was required to achieve the initial cooling performance. Again this was identified as contact of the copper cap in the dewar base. The fall-off in performance when cooling the substrate to 80K at ambient temperature of 20°C and 70°C is shown in Figure 21. While initially it was realised that the copper cap was out of round and over maximum tolerance in one area, no attempt was made to rectify the situation since a production standard cooler would have been checked using a go/no-go gauging system. The area of contact is shown in Figure 22, indicated by an arrow. As before, reduction of the diameter by very careful filing effected a clearance, and re-testing demonstrated that the initial performance results were regained. It could be assumed that during initial testing, there must have been point contact, the area increasing with time due to the ductile nature of the copper cap thus effecting a thermal short.

5.3.2 Serial No. 1108 has undergone acceptance tests and is awaiting life testing when a spare laboratory test station becomes available. The diameter of the copper cap measured 6.78mm, within the revised dimension limits of 6.80mm maximum, 6.75mm minimum. However, excess solder from the copper cap/cold finger tube join had flowed on to small areas of cap making a local maximum size of 6.850mm.

6. DISCUSSIONS AND CONCLUSIONS

- 1) All engines as delivered ran with no mechanical problems, which was itself noteworthy.
- 2) The initial problem of outgassing of water vapour within the Helium charged components, was identified and remedied by the manufacturer.
- 3) Departure from concentricity, orthogonality and accuracy of form of both cold finger and encapsulation have caused recurring deterioration in performance. It is emphasised the clearances are critical, especially when two manufacturers have a policy of remedial liaison. Production cold fingers and production encapsulations now seem to be effectively tolerated, but this is the main area where vigilance is essential.
- 4) An AC engine requires an inverter pack to operate from DC battery supplies in the field. Units now available have efficiencies of approx. 80%.
- 5) The need to keep the warm end of the cold finger within 10 degrees of ambient (by forced draught) must be stressed. Efficient performance, especially at the higher ambients, relies on this.
- 6) It is recognised that testing the engine at an ambient of 70°C with overall heat loads of up to 700mW is unduly demanding. However, the cooling engine arena is littered with models which struggled to meet a reduced specification in infancy and died before maturity. This is not a failing of the Philips Usfa range: they have provided a great deal of data because they survive.

The overall conclusion reached is that the Philips Stirling cycle cooling engine is efficient, durable and available for Service use. Future developments can be expected to optimise what is already a deployable component.

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APPENDIX I - Determination of the heat load imposed on a cooling engine by a detector encapsulation.

The heat load of a detector encapsulation can be calculated from the latent heat of evaporation of LN_2 and LO_2 measuring the minimum flow-rate of gas-evolution from the upright encapsulation dewar bore.

The figure thus obtained does not take into account that the temperature of the boil-off gas at the entrant of the dewar bore is higher than the boiling point of the liquid due to the sensible heat intake from the walls of the dewar bore. Thus the total heat load of the encapsulation would be higher.

In order to determine the encapsulation heat load imposed on the cold finger of a cooling engine, together with the temperature drop between the cold finger and the substrate, the following experimental test schedule was devised.

1. Tests

- 1.1 Determine the heat load of 3 encapsulations by monitoring the gas flow due to the evaporation of LN_2 .
- 1.2 Test in turn the 3 encapsulations with a cooling engine maintaining the same operating conditions.
- 1.3 Determine the revised heat loads of the 3 encapsulations.
- 1.4 Measure the absolute cooling power at the cold finger of the cooling engine.
- 1.5 Compare the absolute cooling power with the results obtained with the 3 encapsulations and the revised heat loads and determine the temperature drops at the interface.

2. Results

- 2.1 The heat load of a detector encapsulation from the observation of the minimum gas flow rate due to the latent heat of evaporation of LN_2 :

Latent heat of evaporation of liquid nitrogen at 77.2K and 1 atmos. pressure

= 199 Joules/gram.

Density of liquid nitrogen at 77.2K and a 1 atmos. pressure

= 808 gms/litre

Latent heat of evaporation = 199×808 Joules/litre LN_2

1 Litre of LN_2 at 77.2K and 1 atmos. pressure yields

696 litres gas at 300K and 1 atmos. pressure.

Latent heat = $\frac{199 \times 808 \text{ Joules/litre gas at 300K.}}{696}$

Equivalent heat load

$$= F \times \frac{199 \times 808}{696 \times 60} \text{ Watts}$$

$$= 3.849F \text{ Watts}$$

Where F is the gas evolution rate in litres per min. (or c.c.s/min. to give result in mWatts) as measured by a Rotameter flowmeter piped to the dewar bore.

The heat loads of 3 engine compatible detector encapsulations were determined by this method. The encapsulations comprised of one used extensively for the tests in the main report and two others with a somewhat higher heat load due to their type but mainly because of a poor vacuum.

No.	Type No.	Ser. No.	Encapsulation	Flow Rate cc/min.	Heat Load mW
I	T1915	85G00S	R185	27	104
II	T1515	001	R170	65	250
III	T1515	003	R170	91	350

2.2 Cooling engine tested with the 3 encapsulations

Encapsulation I was assembled on to the cold finger of the Philips UA7041, SN 0414 cooling engine. A thin smear of thermal transfer grease supplied with the cooling engine was applied to the pad on the thermal coupling on the cold finger. The sealing-ring for the cold finger flange was given a smear of silicone high vacuum grease, the encapsulation was carefully located over the cold finger and held in place with three-equispaced clamps.

The temperature of the array was monitored by a sensor diode powered by a 1mA constant current source.

The encapsulation was cooled to achieve the same voltage reading at the sensor diode as when it was cooled by liquid nitrogen. Bias loads at 50mW increments were applied, monitoring the input power requirement to maintain the initial cooled substrate temperature.

This was repeated for encapsulations II and III maintaining the same operating conditions at 20°C ambient and the same air flow rate from the cooling fans. The results obtained are shown in Fig. A1.

2.3 Revised heat loads of the detector encapsulations

Ignoring any small heat convection losses between the cold finger and dewar bore, the heat load at the cold finger is a summation of the encapsulation heat load and the bias load applied to the array, therefore, the vertical separation between the graphs plotted must be due to the differences of the heat loads of the encapsulations.

The differences between the heat loads:

Encapsulations I and II was 220 mW
Encapsulations I and III was 375 mW

This can be graphically demonstrated by offsetting the results obtained using encapsulations II and III by these differences Fig. A2.

The heat loads obtained from the evaporation rate of LN₂ gave differences of:

Encapsulations II-I = 250 - 104 = 146 mW
Encapsulations III-I = 350 - 104 = 246 mW

Comparing the 2 sets of differences

$$\frac{\text{Cooling Engine Heat Load}}{\text{Evaporation LN}_2 \text{ Heat Load}} = \frac{\text{II-I}}{\text{II-I}} = \frac{220}{146} = 1.507$$

$$\frac{\text{III-I}}{\text{III-I}} = \frac{375}{246} = 1.524$$

Mean difference of 1.515

This is the modifying factor that needs to be applied to the heat load figure obtained from the evaporation rate of LN₂ to give a realistic heat load of the encapsulation on the cold finger.

Revised heat load

Encapsulation I	104 x 1.515	=	157 mW
" " II	250 x 1.515	=	379 mW
" " III	350 x 1.515	=	530 mW

2.4 Absolute cooling power of the cooling engine

The absolute cooling power at the thermal coupling of the cold finger was measured by removing the detector encapsulation and mounting a calibrated thermocouple and heater to a copper platform which had been soft soldered to the pad on the thermal coupling. The cold finger was loosely wrapped with several layers of Mylar film in order to reduce radiation heat intake. A vacuum dome was installed over the cold finger, and continually pumped by a turbo molecular vacuum pump during the tests to eliminate heat intake due to air convection/conduction.

Power was applied to the cooling engine and was monitored when 70K was achieved at the thermal coupling pad. Heating load was applied at 50 mW increments, monitoring the increased input power required to maintain the cryogenic temperature. The test was repeated at 77.2K (LN₂), 80K, 90K and 100K temperatures. Allowances were made for the thermal conduction losses through the leads to the thermocouple and heater and the results were plotted Fig. A3.

2.5 The total heat load imposed on the cold finger by the detector encapsulation when biased is obtained by summing the revised heat load to the initial test results. When this load curve is superimposed on the absolute cooling performance, it can be seen to lie between the 77.2K and 70K isotherms, Fig. A4. Its line must represent the temperature at the pad of the thermal coupling. This temperature was determined for encapsulation I, since encapsulations II and III were not representative of typical heat loads encountered in service. Re-plotting cold finger load against temperature for input powers of 10, 20, 30 and 40 W is shown in Fig. A5. A best-fit line is drawn through the intersection of the heat load of encapsulation I plus bias loads and their input power requirements to maintain the substrate at the LN₂ cooled temperature.

The temperature difference between liquid nitrogen and the array on the substrate platform is quoted (Mullard figure) typically at 1.6K.

The temperature difference between the thermal coupling pad and LN₂ temperature was 2.5K for a detector encapsulation with zero bias, rising to 6.5K for 500 mW bias load.

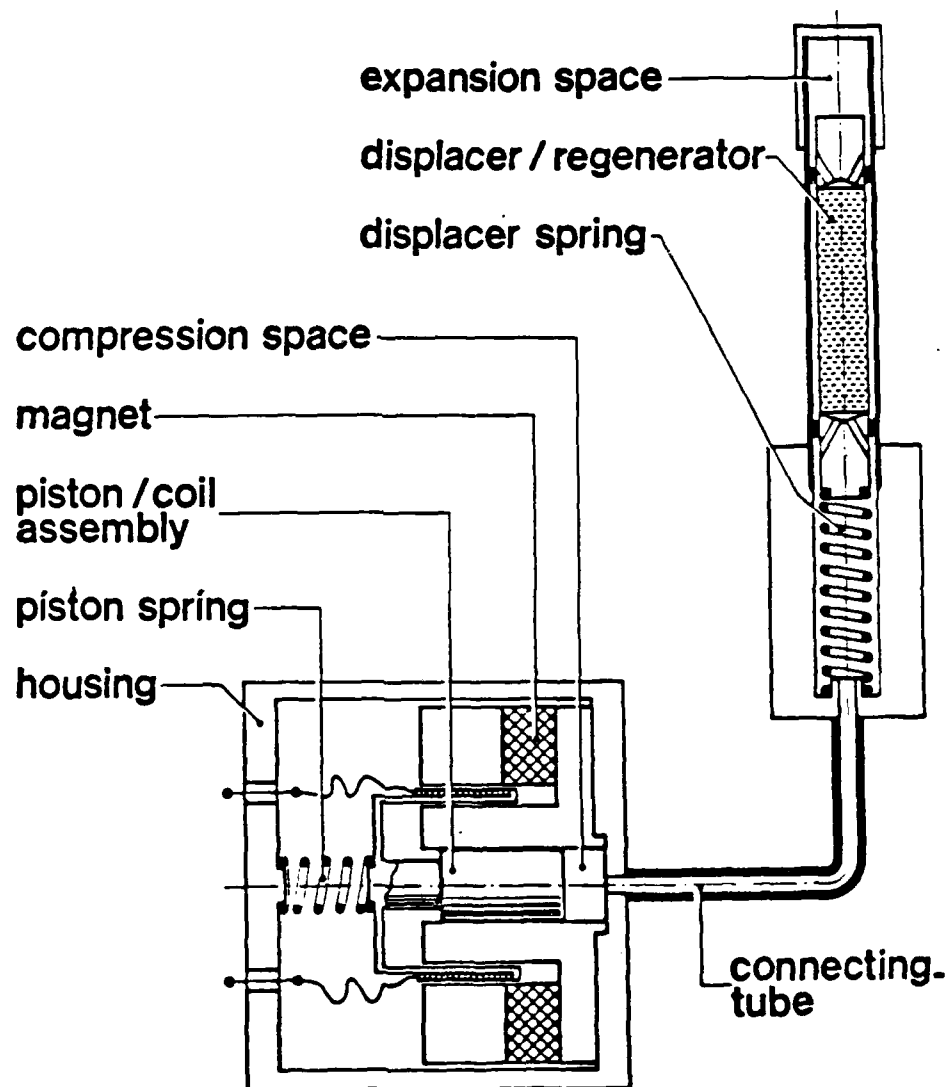
Conclusions

In order to relate the performance of a cooling engine quoted in absolute terms to that when cooling a typical detector encapsulation at normal ambient temperature conditions, the following generalisations may be safely used.

- 1) The heat load imposed on the cold finger by the detector encapsulation is 1.5 times that figure obtained by the LN₂ boil off method.
- 2) For a photoconductive array operating with 100 mW bias, a temperature difference of 5K can be assumed between the array and the pad of the thermal coupling.

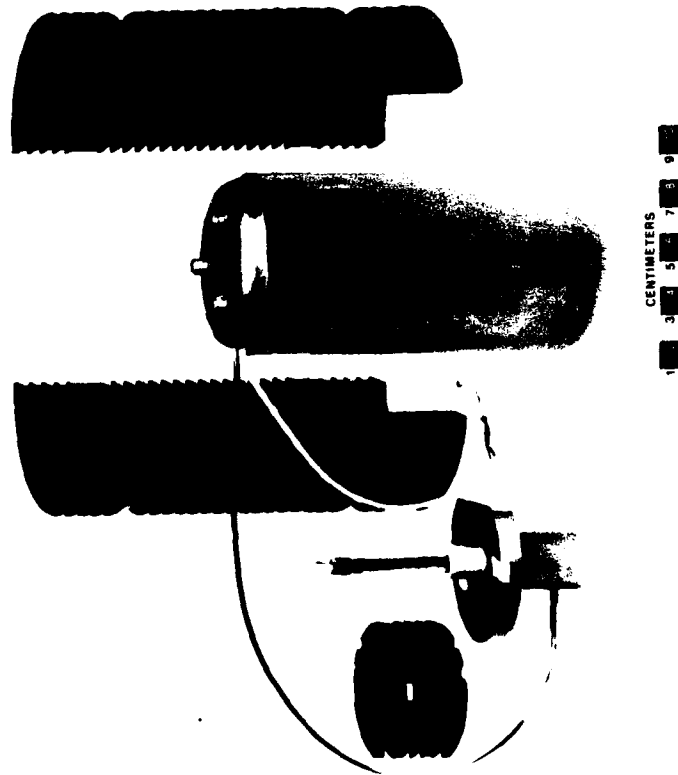
An adequately designed thermal coupling is required and the necessary precaution on assembly must be taken.

FIG. 1



Diagrammatic cross section of the split
Stirling refrigerator with electrodynamic drive

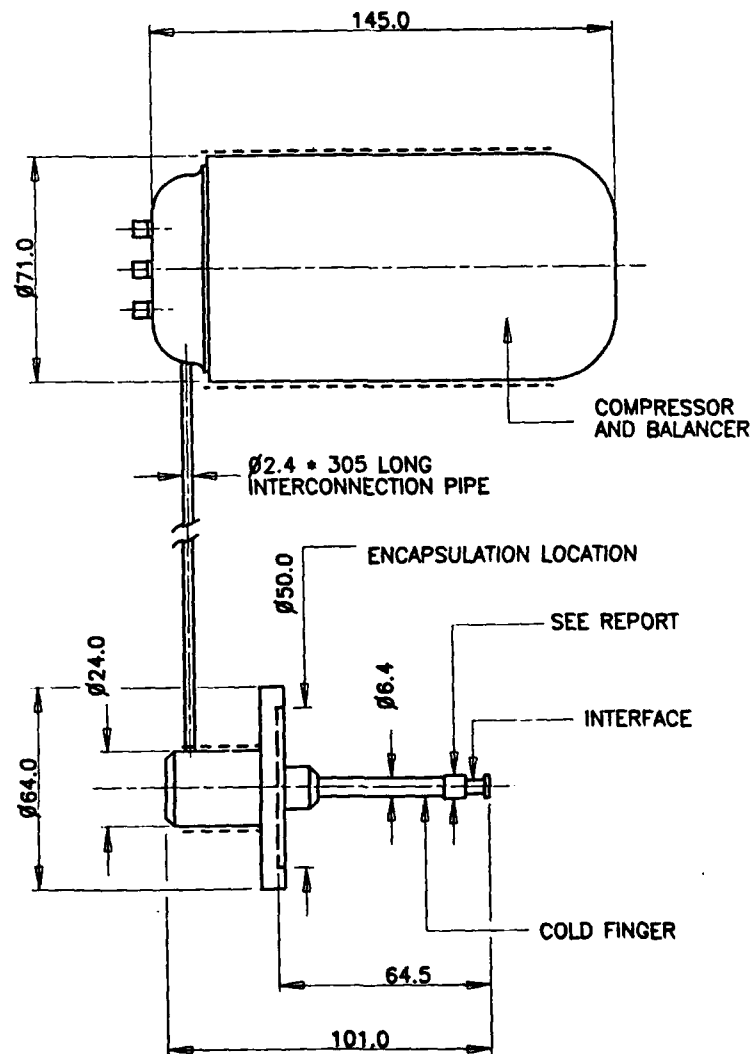
FIG. 2



COOLING ENGINE UA 7041 AND HEAT SINKS

OUTLINE DRAWING GIVING KEY DIMENSIONS
CRYOGENIC ENGINE UA 7041/00
MODEL WITH INTEGRAL BALANCER

FIG 3a

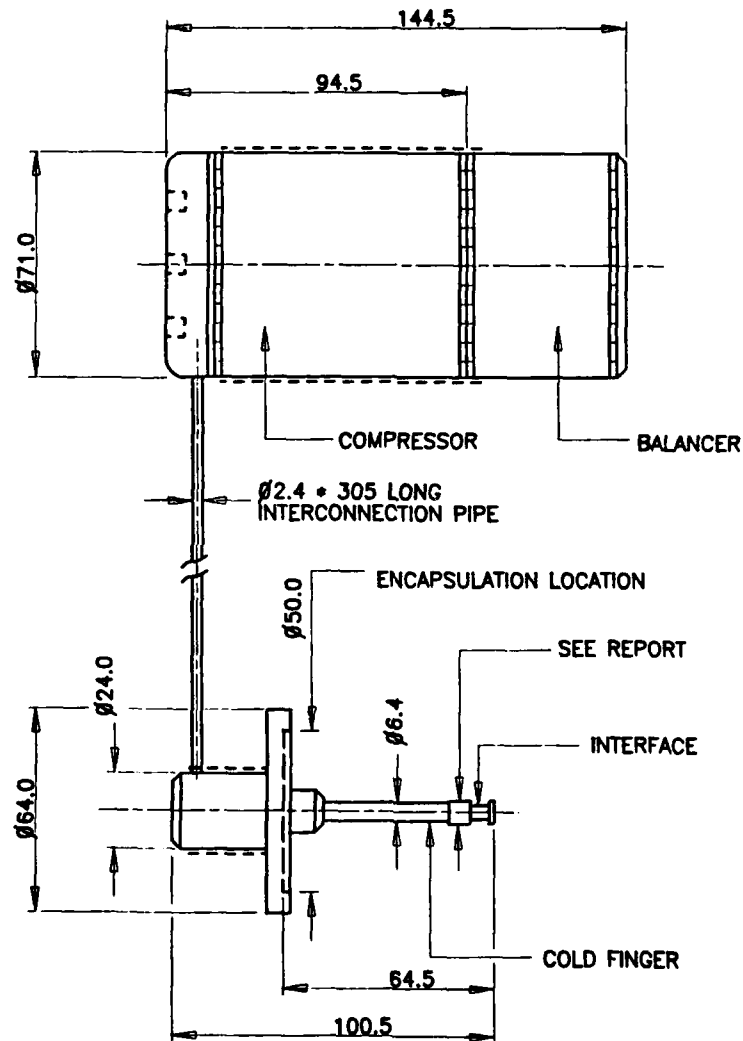


HEAT SINKS TO BE FITTED
TO SURFACES INDICATED -----

DIMS IN mm
WEIGHT 1988 gms.

OUTLINE DRAWING GIVING KEY DIMENSIONS
 CRYOGENIC ENGINE UA 7041/00
 LATEST MODEL WITH DEMOUNTABLE BALANCER

FIG 3b



HEAT SINKS TO BE FITTED
 TO SURFACES INDICATED -----

DIMS IN mm
 WEIGHT 2095 gms.

FIG. 4

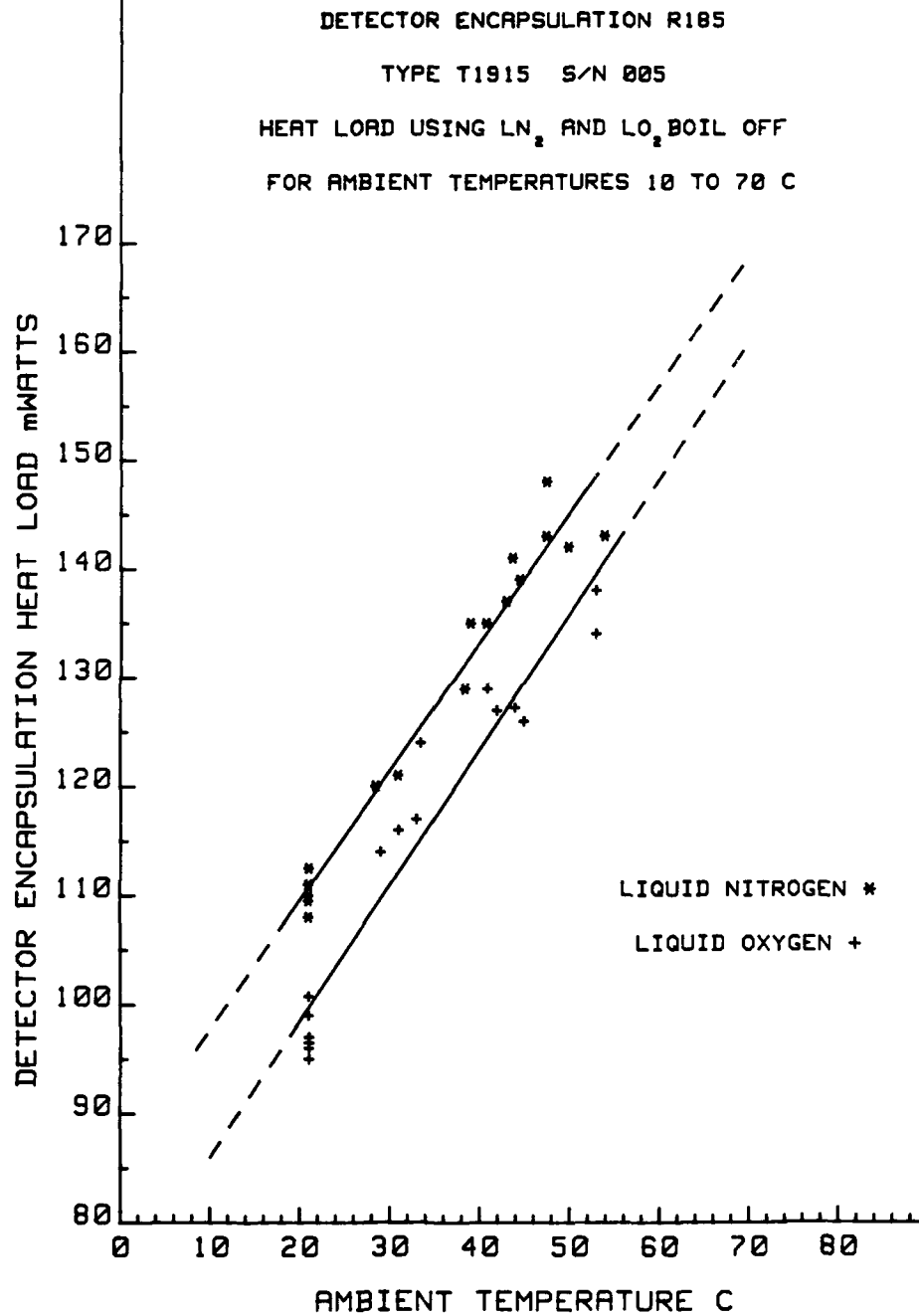


FIG. 5

DETECTOR ENCAPSULATION R185

TYPE T1915 S/N 005

SUBSTRATE TEMPERATURE vs DIODE VOLTAGE

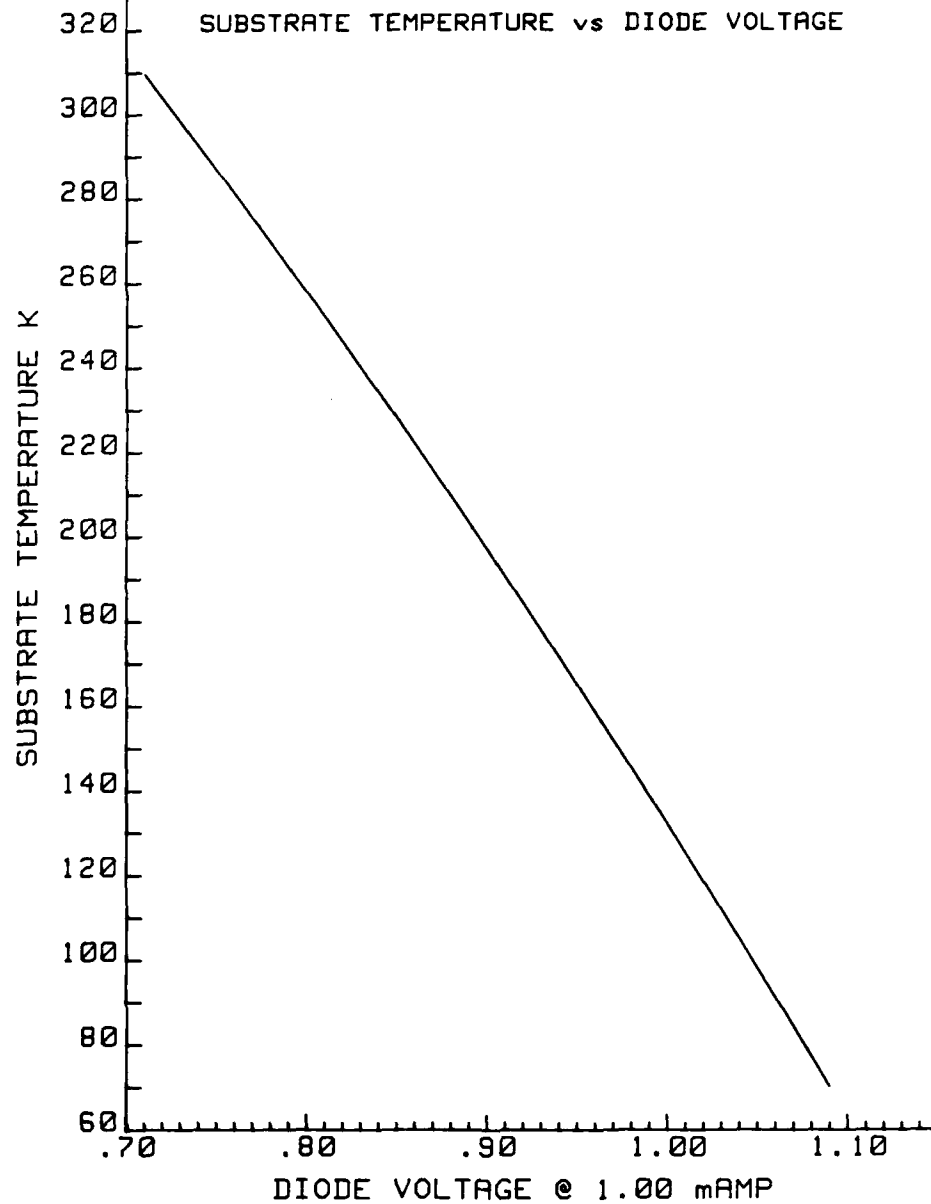
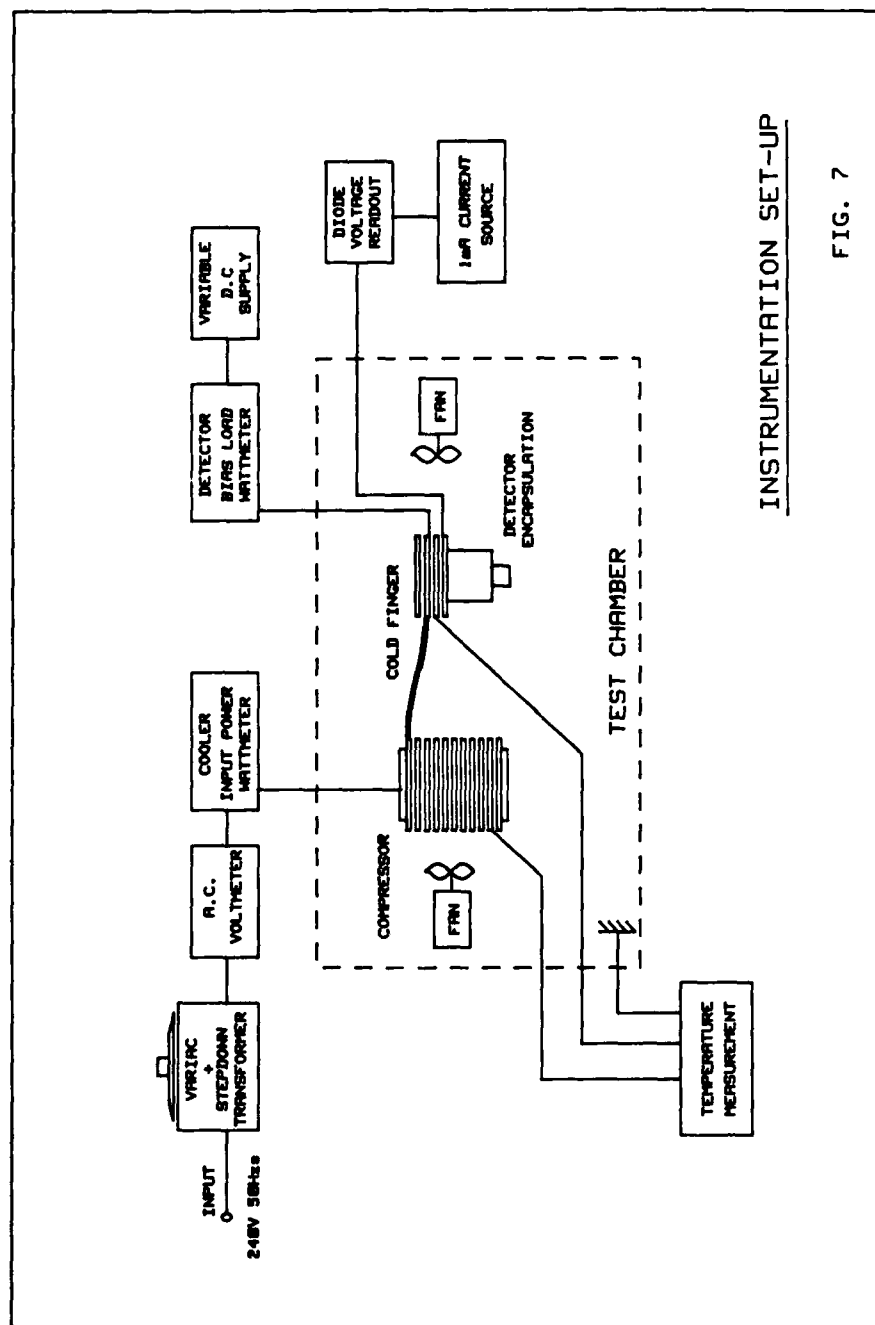




FIG. 6

COOLING ENGINE UA 7041 WITH DETECTOR ENCAPSULATION
AND HEAT SINKS FITTED - MOUNTED ON BASE PLATE



INSTRUMENTATION SET-UP

FIG. 7

FIG. 8

INPUT VOLTS @ AMPS vs INPUT POWER WATTS
AT 20 & 70 C AMBIENT TEMPERATURES

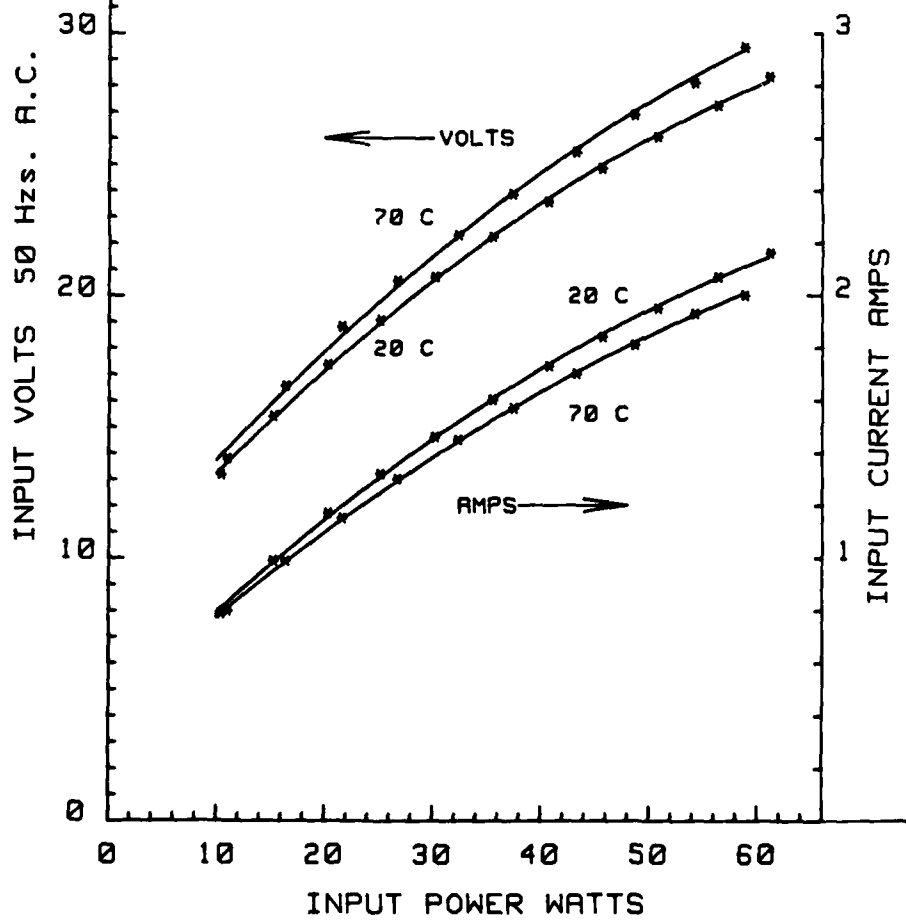


FIG. 9

SUBSTRATE TEMPERATURE K vs COOLDOWN TIME
FOR CONSTANT INPUT POWERS AT 20 C AMBIENT

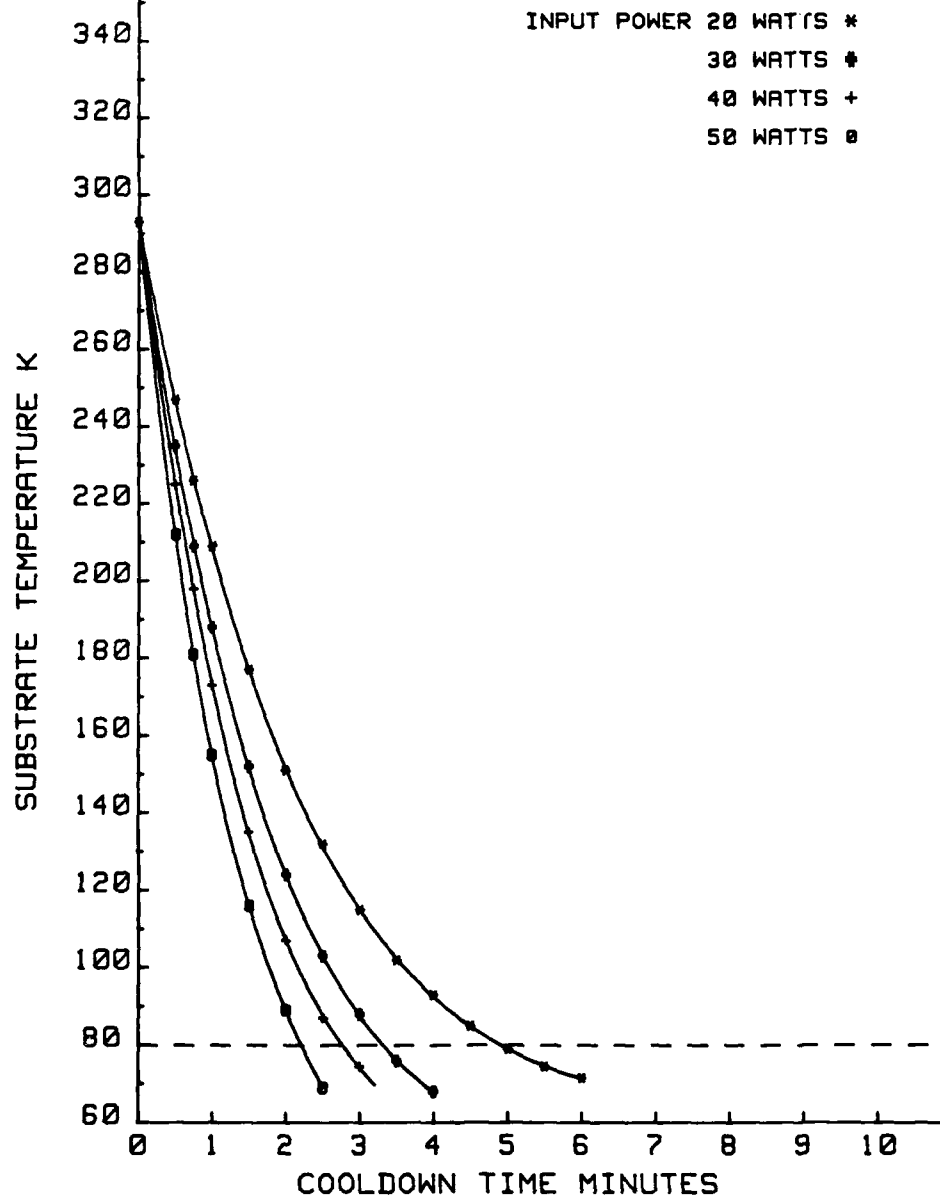


FIG. 10

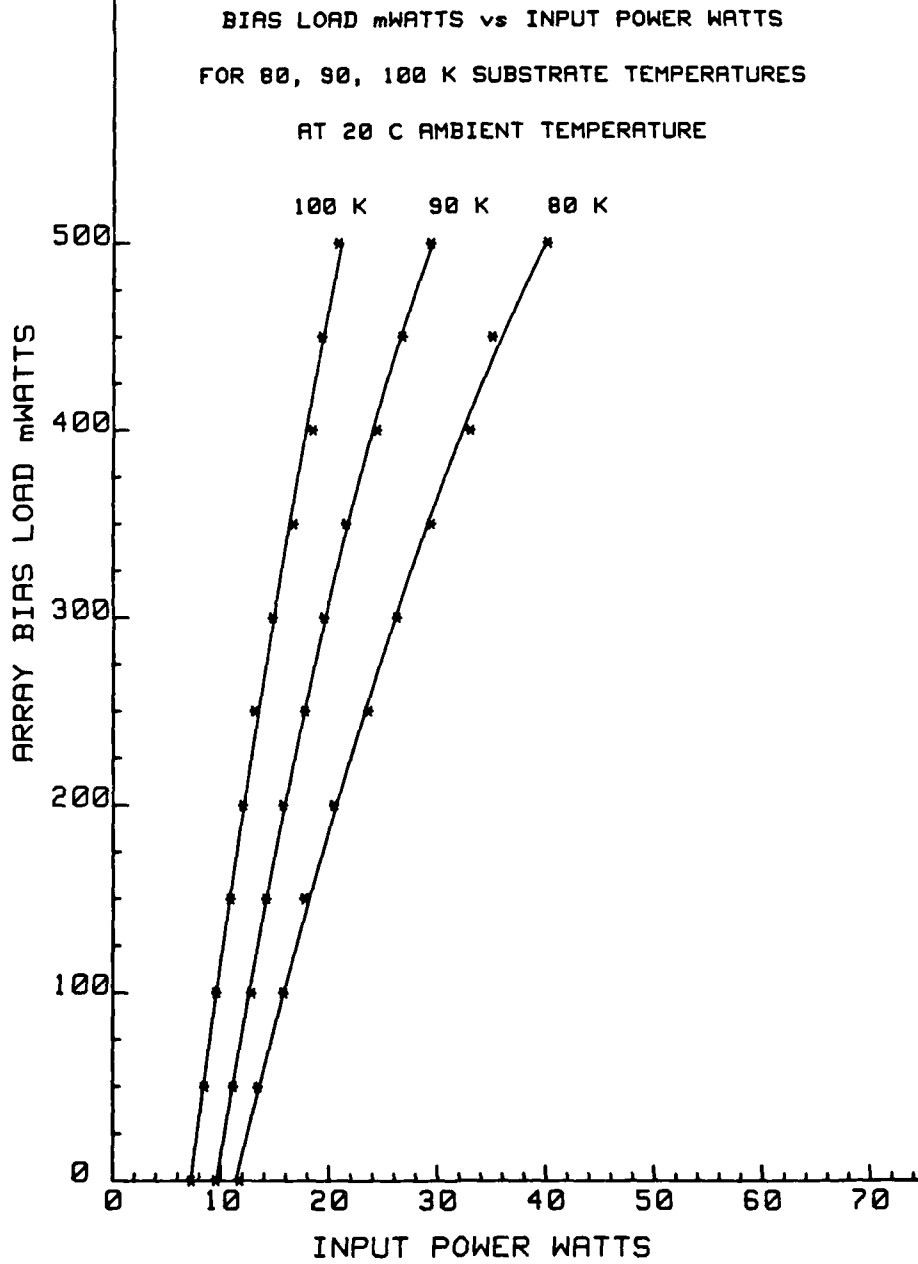


FIG. 11

SUBSTRATE TEMPERATURE K vs COOLDOWN TIME
FOR CONSTANT INPUT POWERS AT 30 C AMBIENT

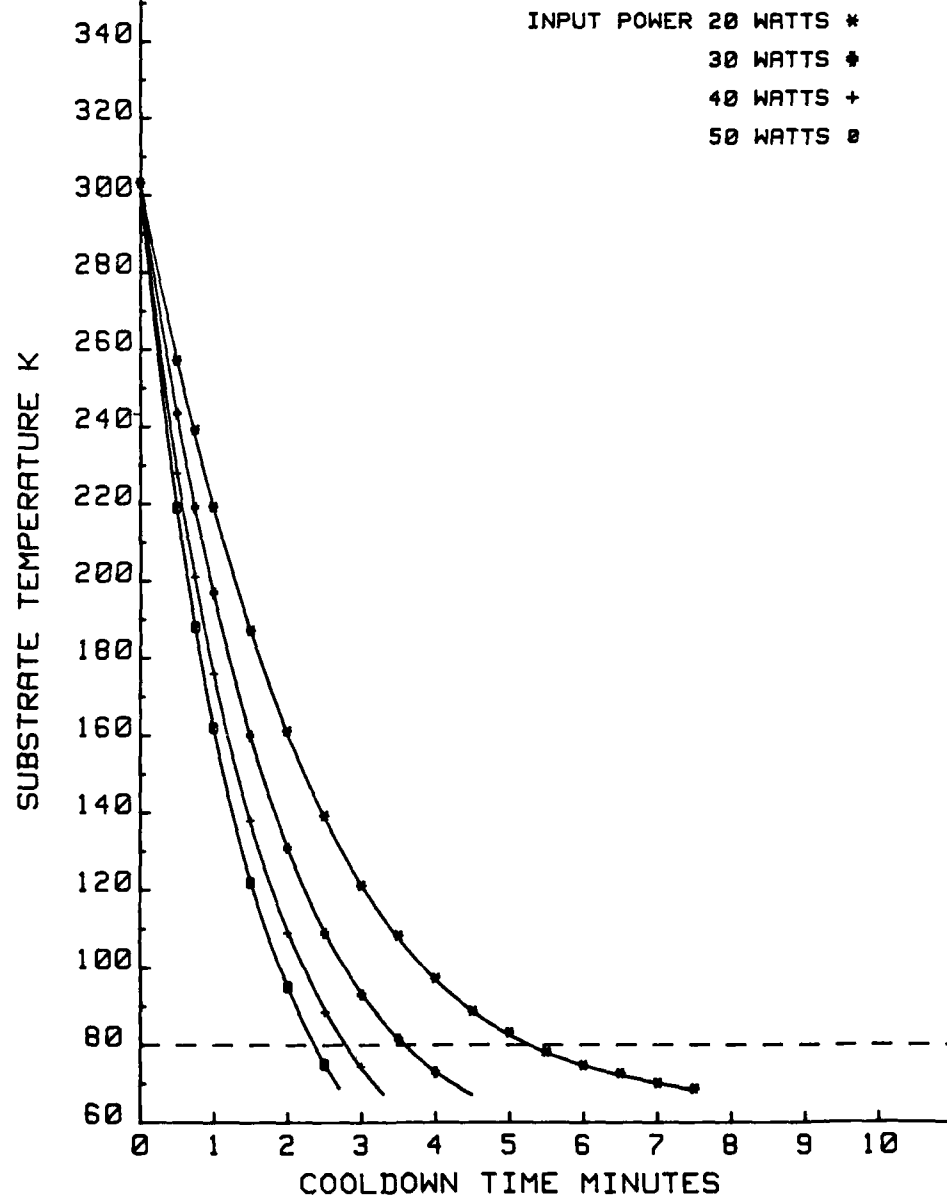
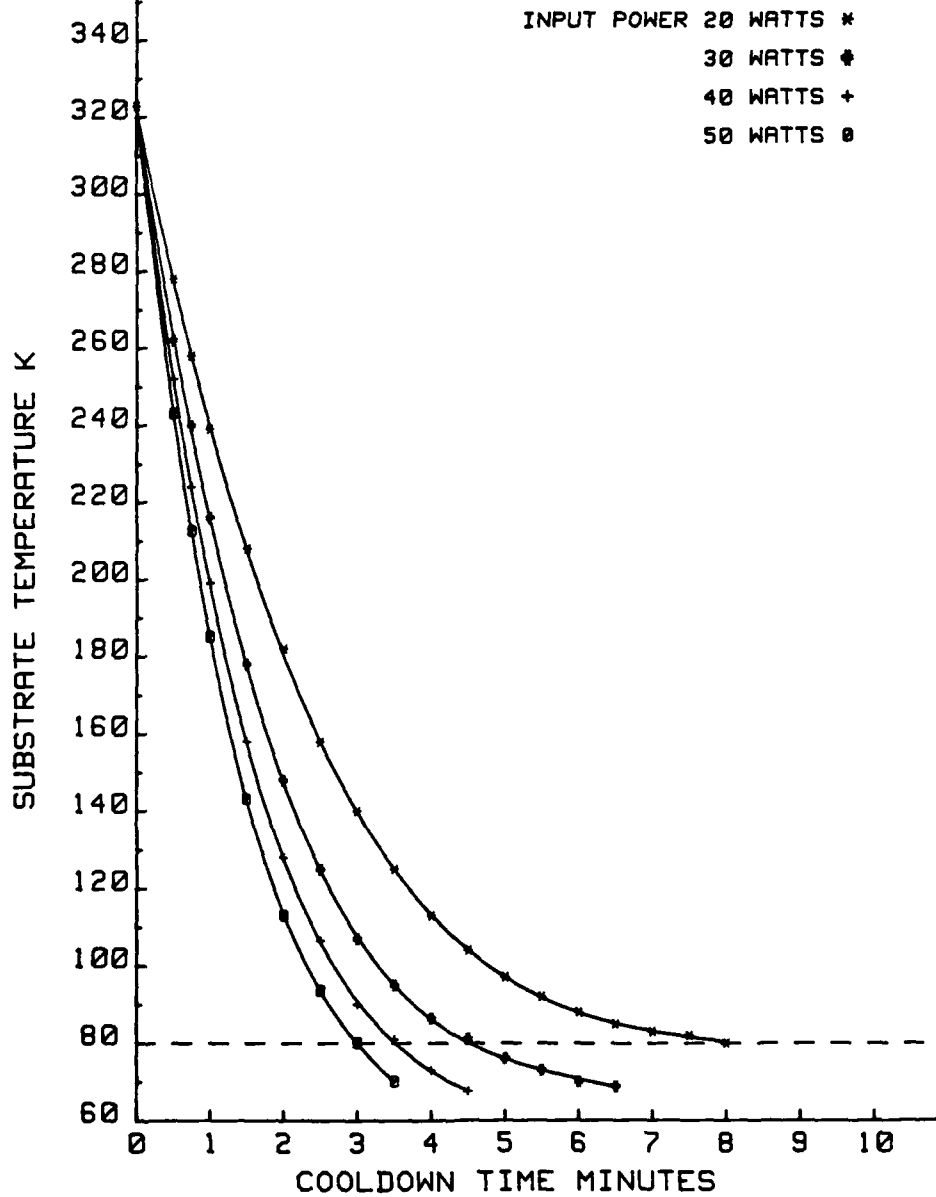


FIG. 12

SUBSTRATE TEMPERATURE K vs COOLDOWN TIME
FOR CONSTANT INPUT POWERS AT 50 C AMBIENT



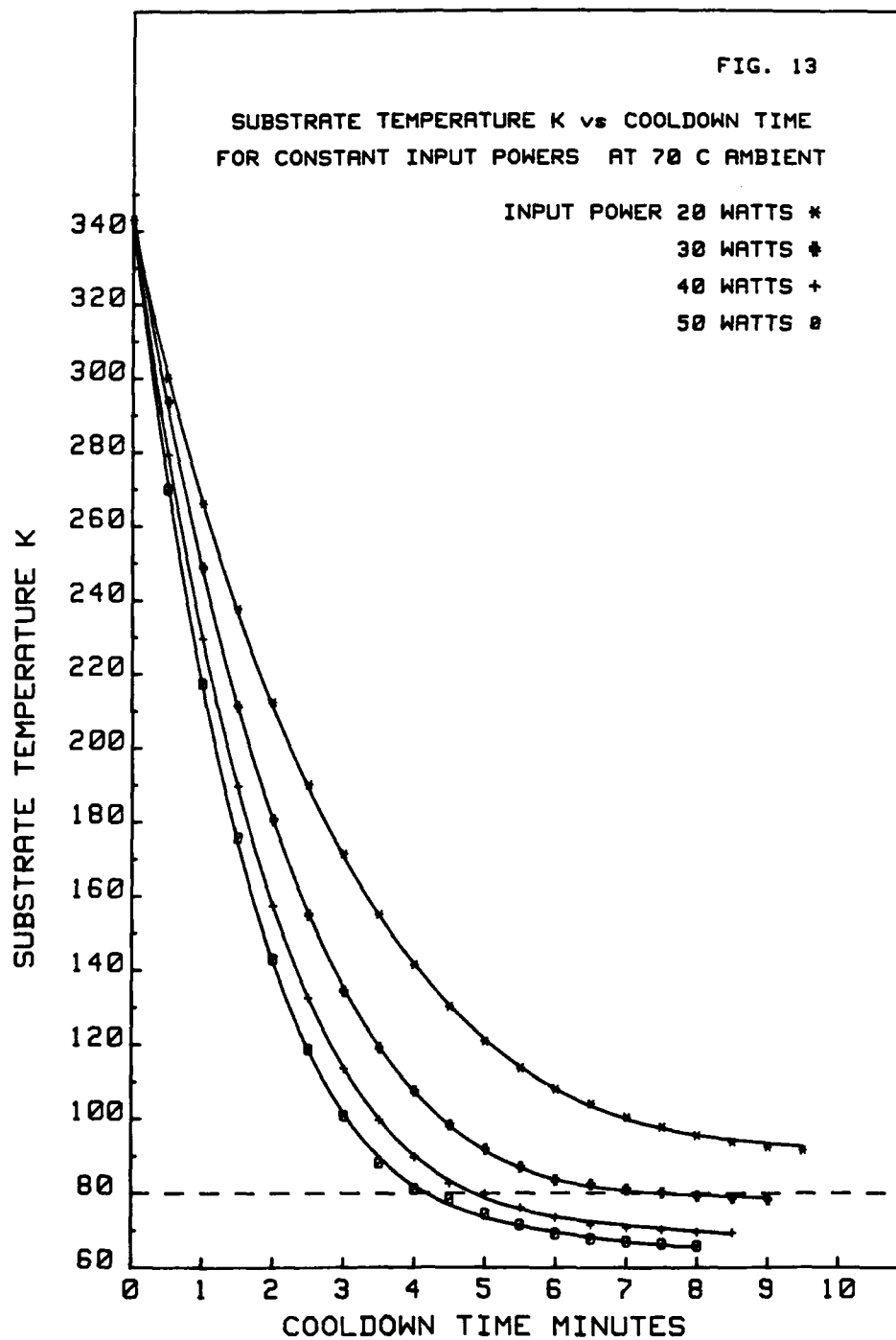


FIG. 14

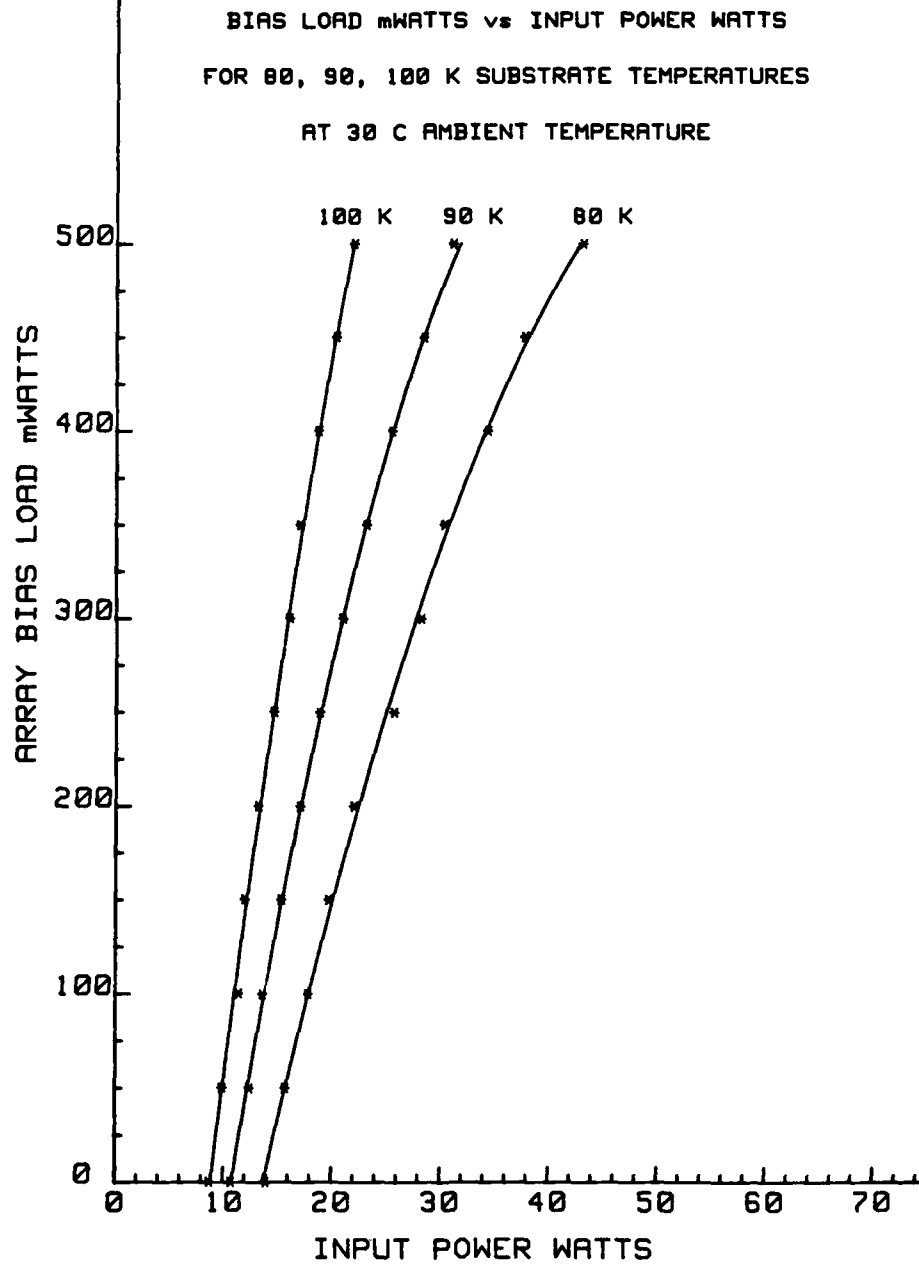


FIG. 15

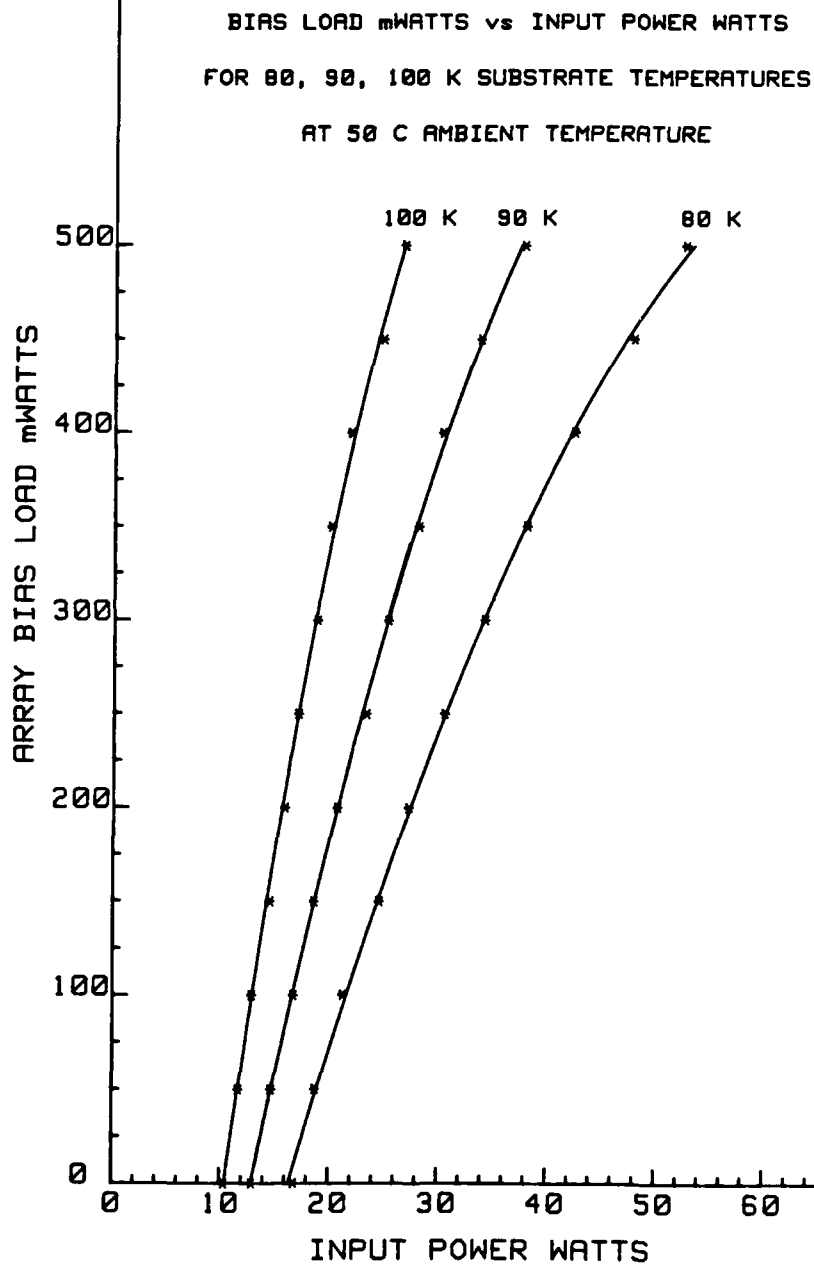


FIG. 16

BIAS LOAD mWATTS vs INPUT POWER WATTS
FOR 80, 90, 100 K SUBSTRATE TEMPERATURES
AT 70 C AMBIENT TEMPERATURE

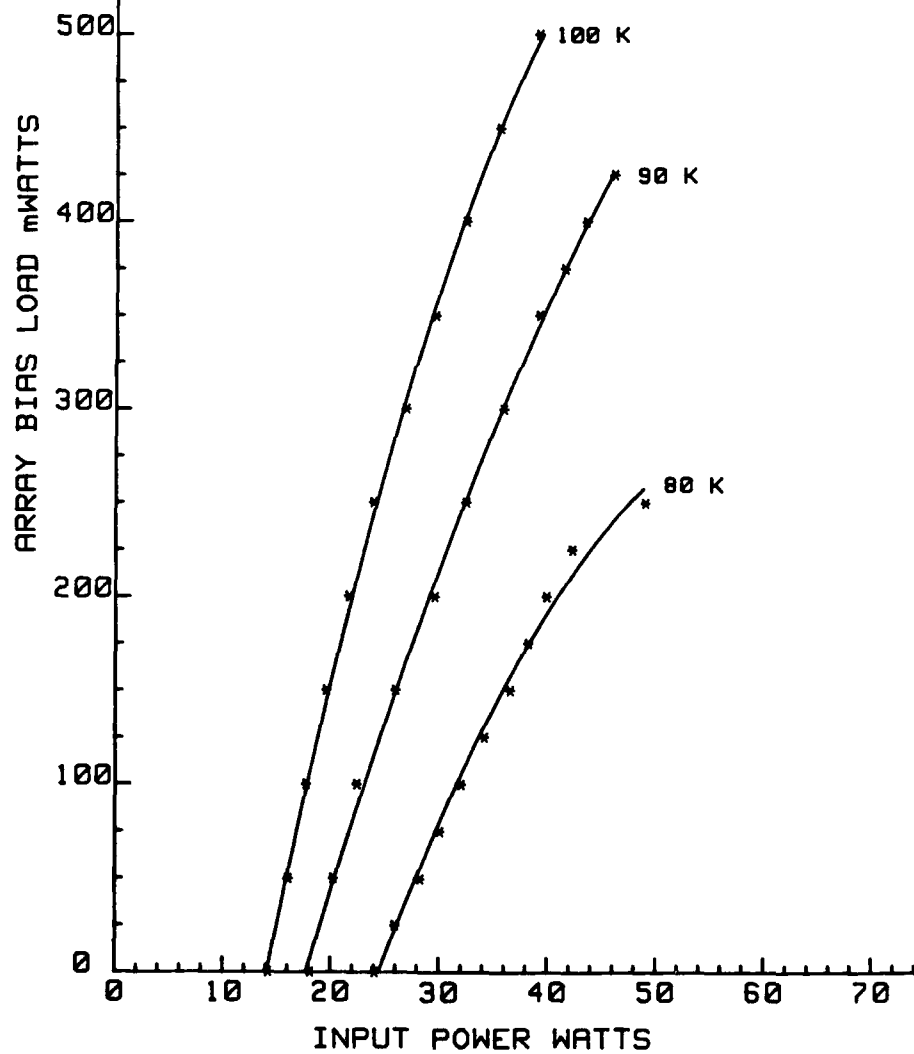
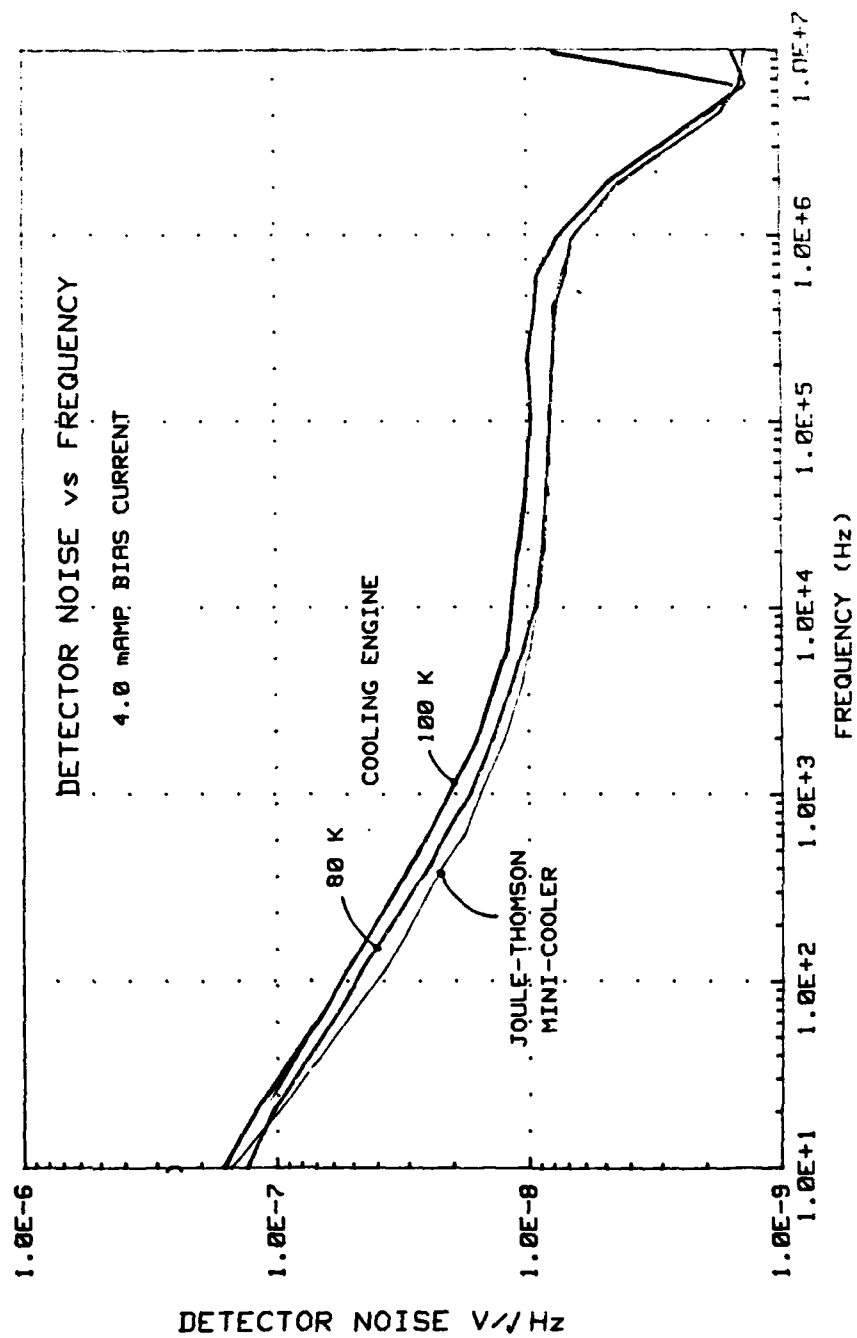


FIG. 17



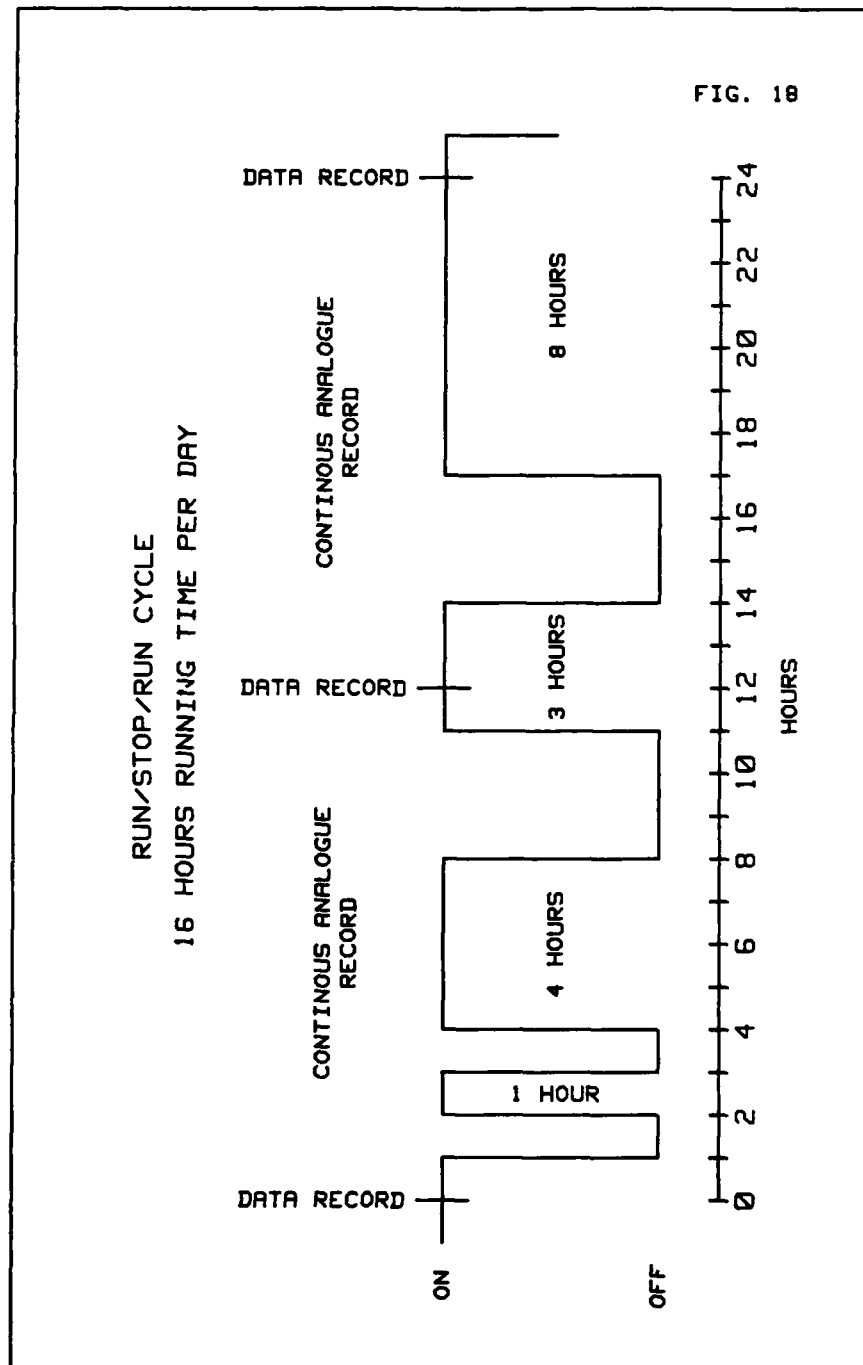


FIG. 18

FIG. 19

PHILIPS UA 7041 S.N. 068 COOLING ENGINE
(GROSS HOURS INDICATED)

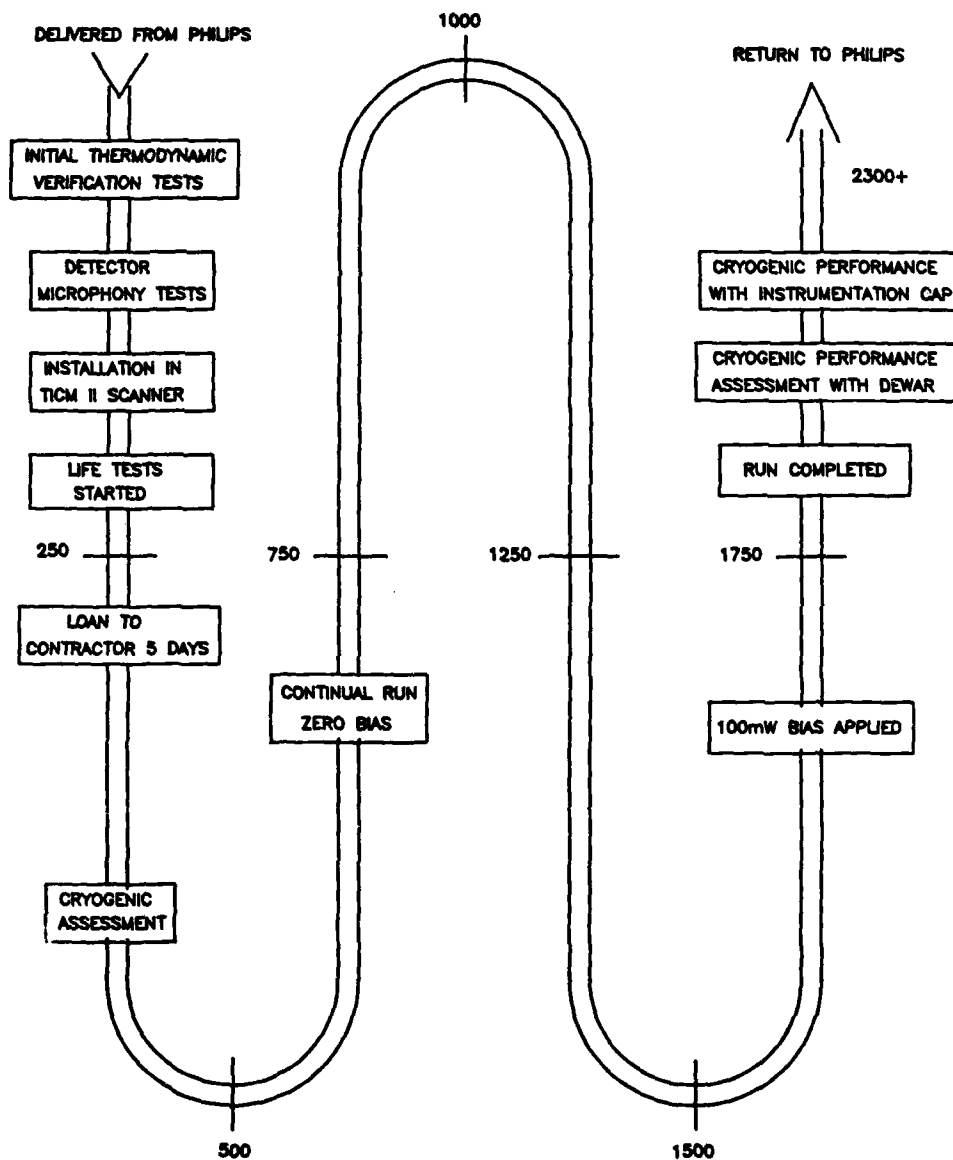


FIG. 20

PHILIPS UA 7041 S.N. 0108 COOLING ENGINE
(GROSS HOURS INDICATED)

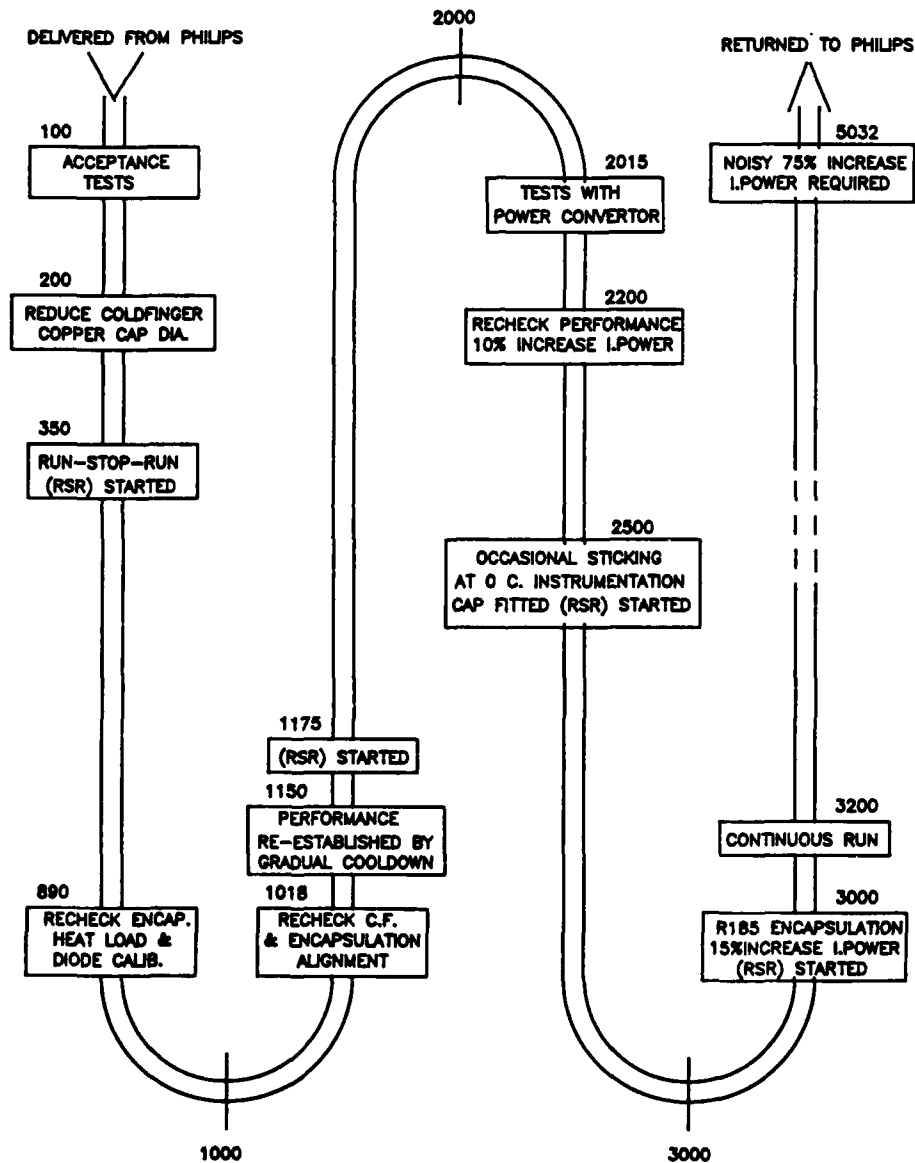


FIG. 21

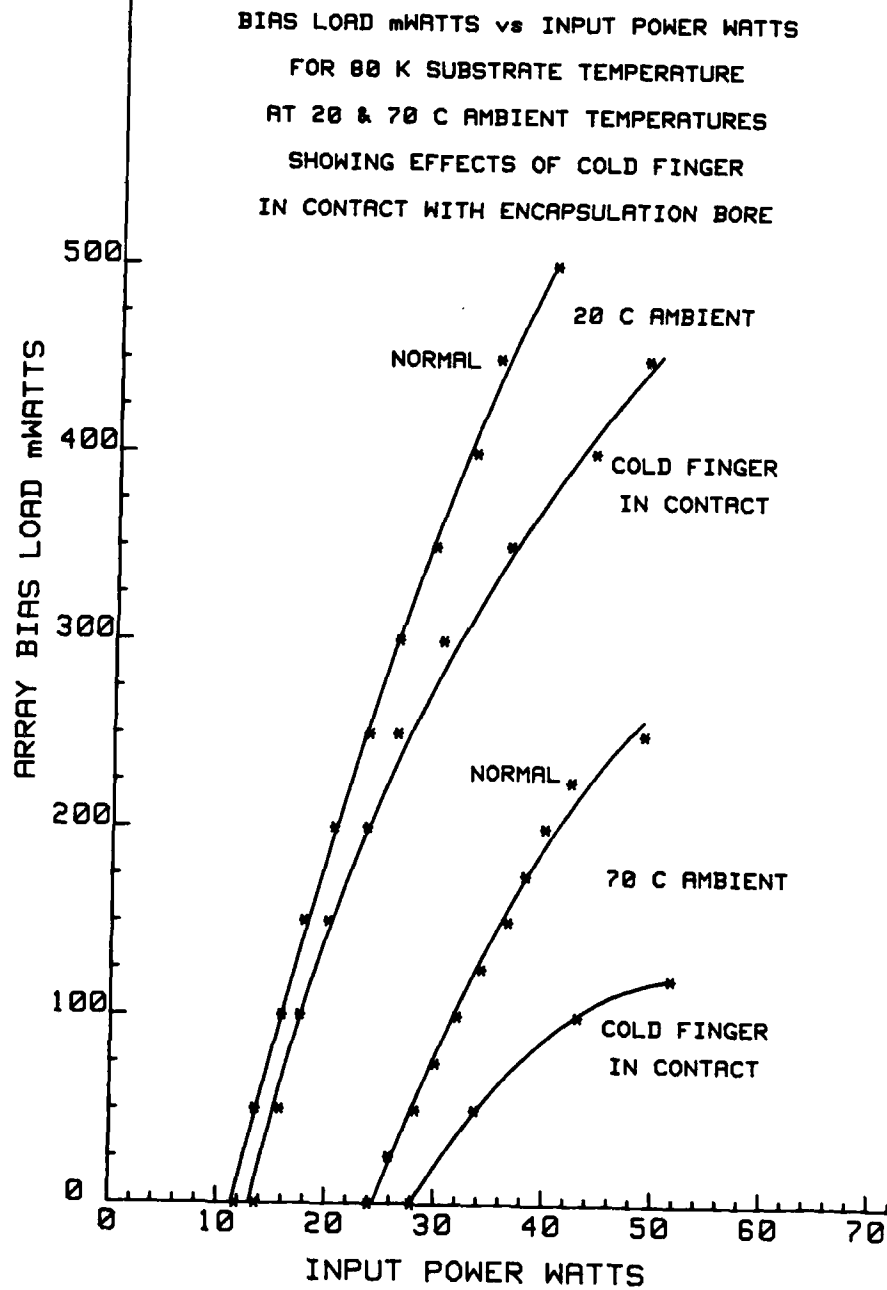
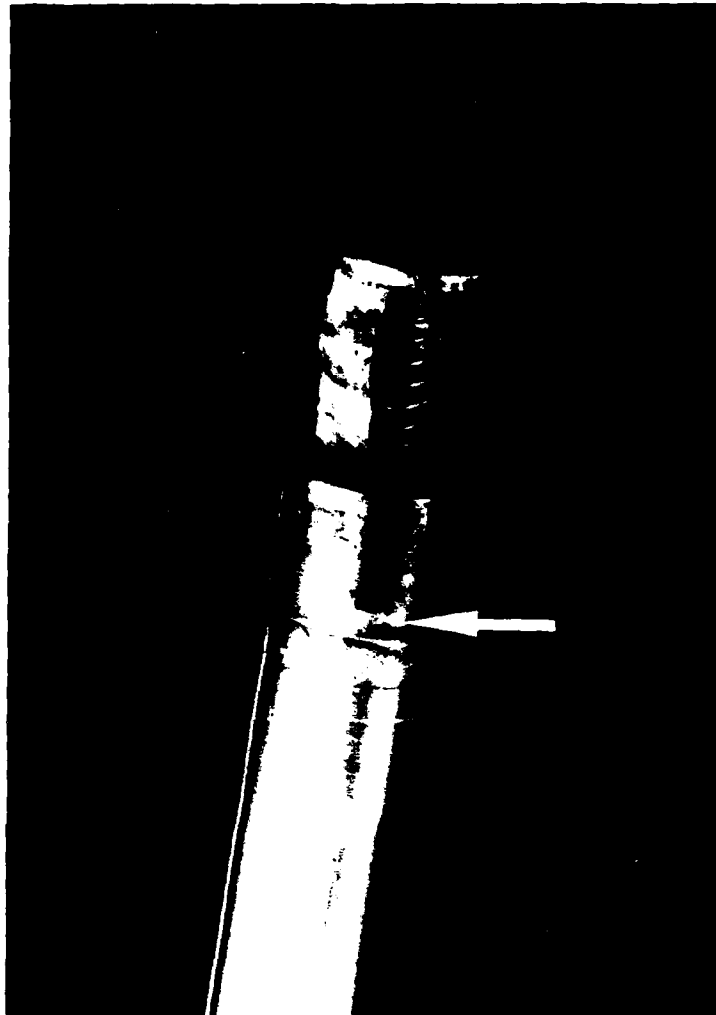


FIG. 22



COOLING ENGINE UA 7041 COLD FINGER
SHOWING AREA OF CONTACT WITH DEWAR BORE

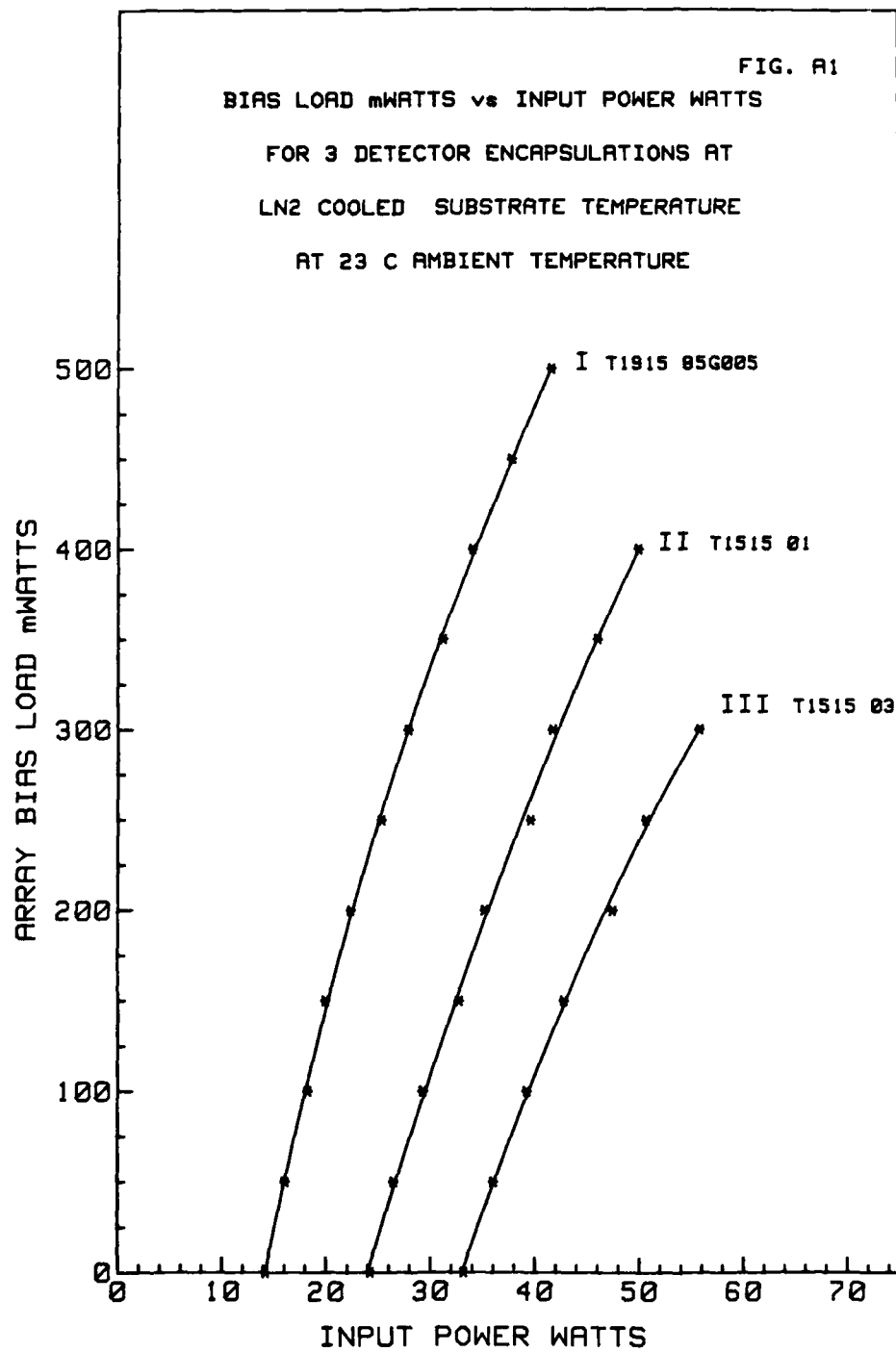


FIG. h2

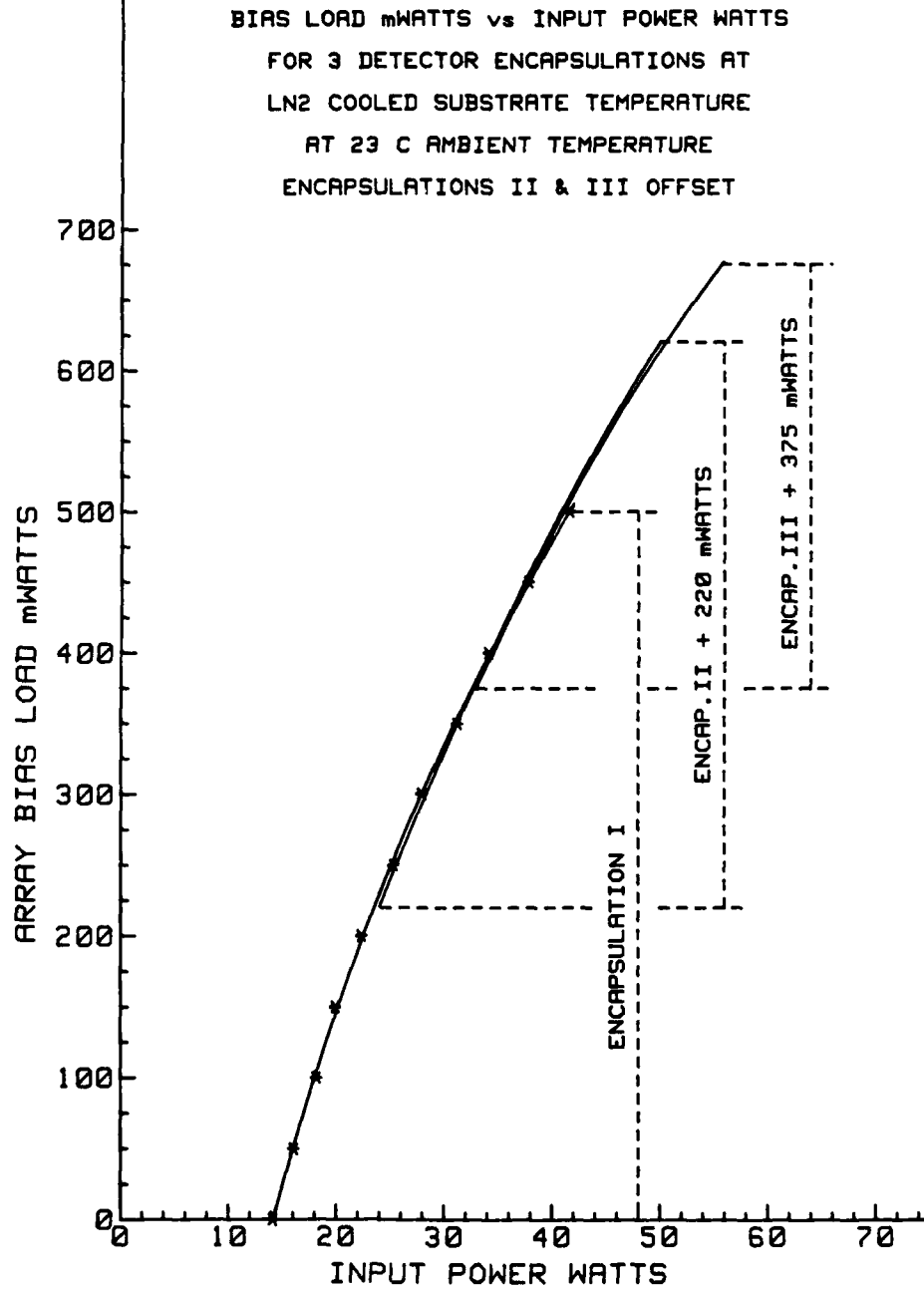
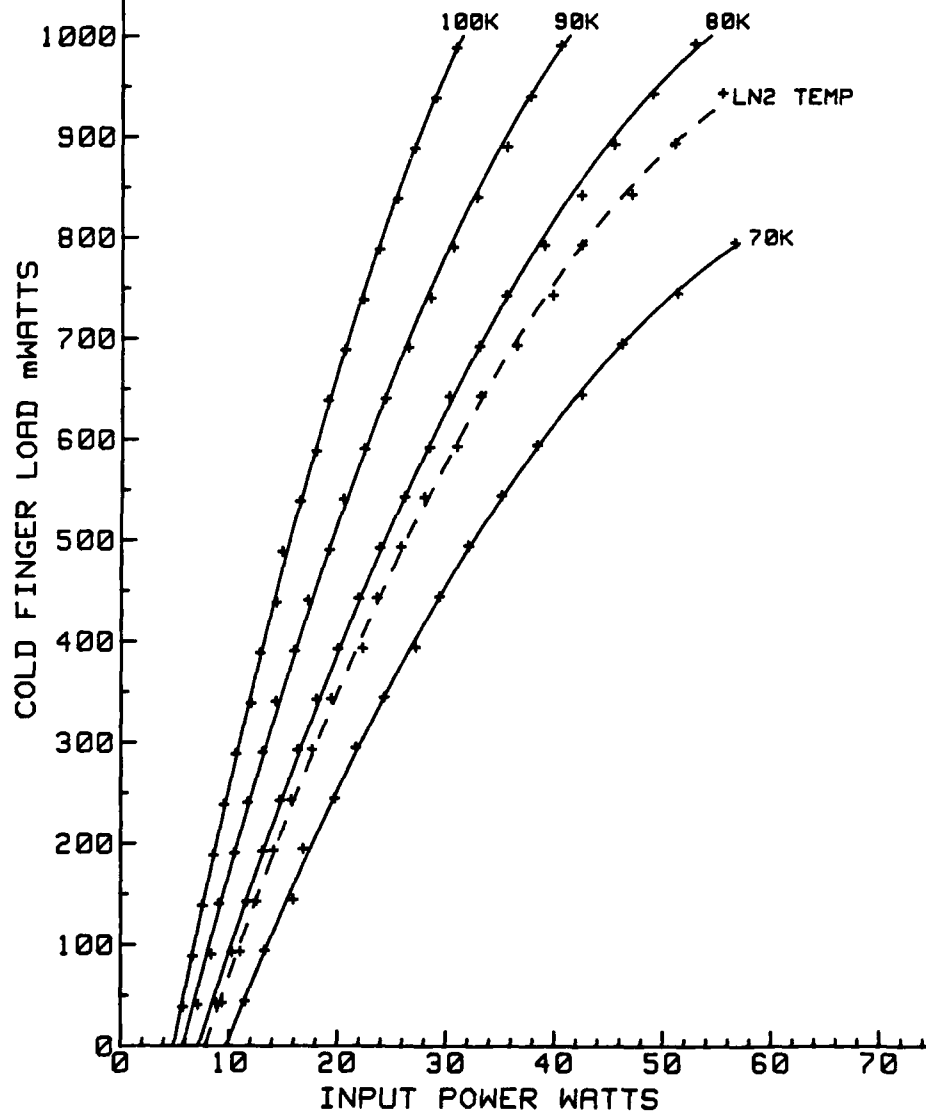


FIG. A3

COLD FINGER LOAD mWATTS vs INPUT POWER WATTS
FOR 70, LN2, 80, 90, 100 K COLD TIP TEMPERATURES
AT 23 C AMBIENT TEMPERATURE



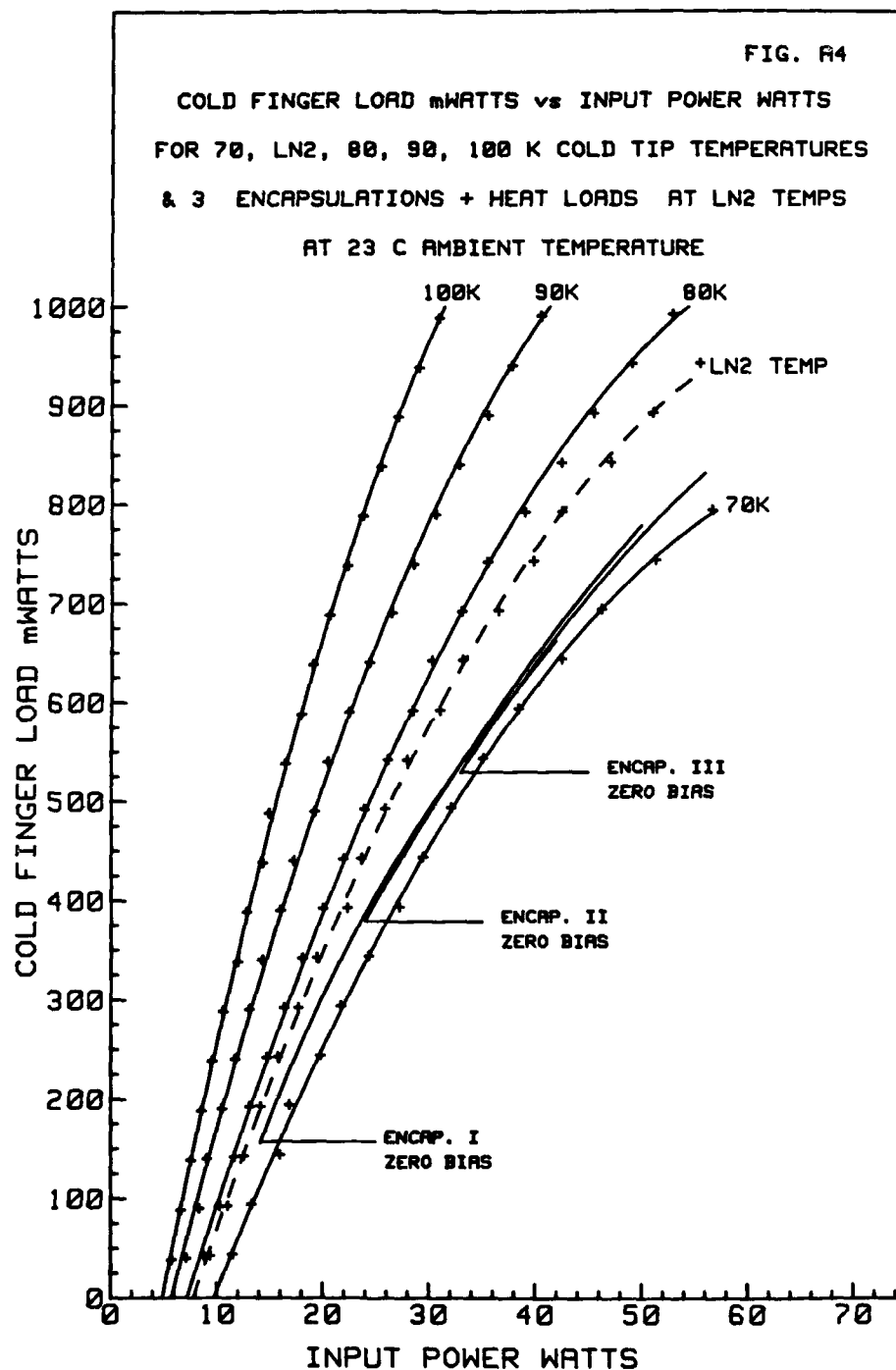


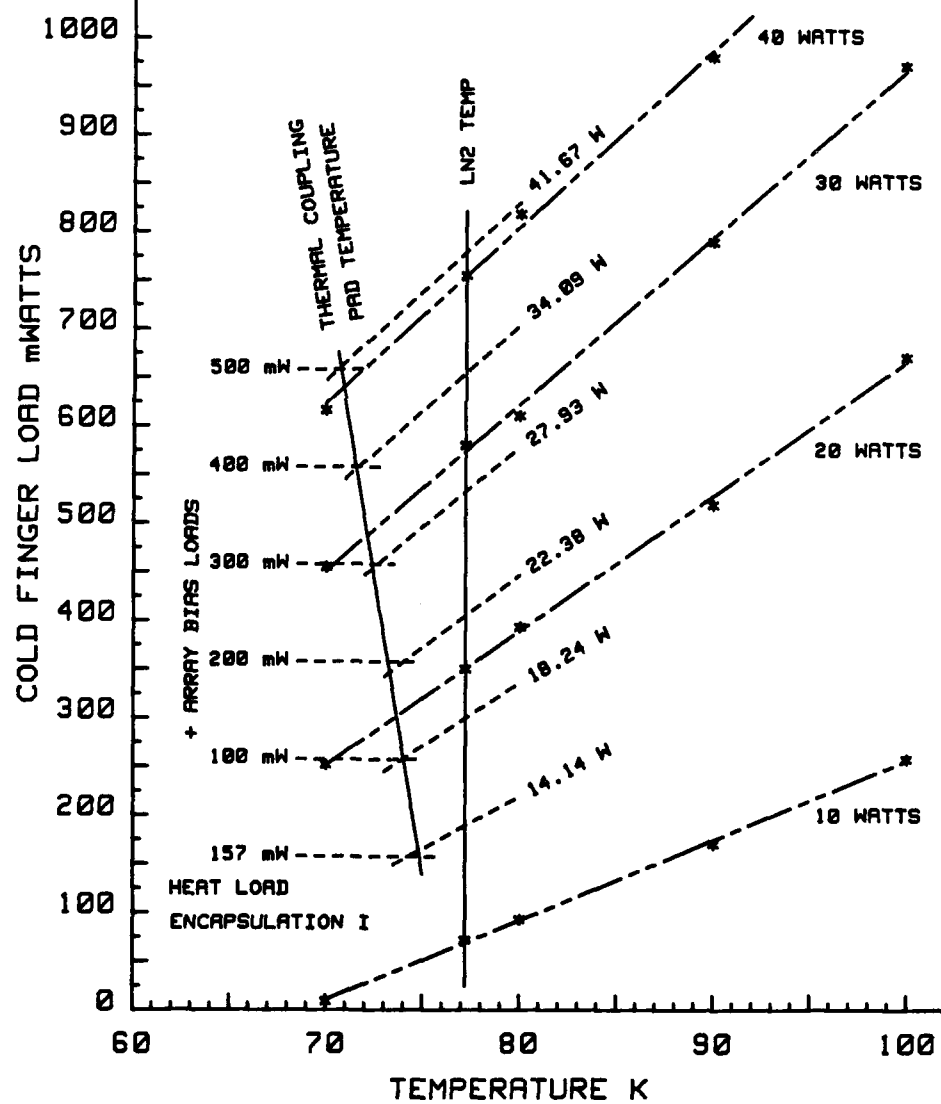
FIG. A5

COLD FINGER LOAD mWATTS vs TEMPERATURE K

FOR 10, 20, 30, 40 WATTS INPUT POWER

+ DETECTOR ENCAPSULATION I

AT 23 C AMBIENT TEMPERATURE



DOCUMENT CONTROL SHEET

Overall security classification of sheet ... UNCLASSIFIED

(As far as possible this sheet should contain only unclassified information. If it is necessary to enter classified information, the box concerned must be marked to indicate the classification eg (R) (C) or (S))

1. DRIC Reference (if known)	2. Originator's Reference Report 87018	3. Agency Reference	4. Report Security Classification Unclassified	
5. Originator's Code (if known) 778400	6. Originator (Corporate Author) Name and Location Royal Signals and Radar Establishment St Andrews Road, Malvern, Worcestershire WR14 3PS			
5a. Sponsoring Agency's Code (if known)	6a. Sponsoring Agency (Contract Authority) Name and Location			
7. Title ASSESSMENT OF THE PHILIPS USFA BV 7 mm COOLING ENGINE				
7a. Title in Foreign Language (in the case of translations)				
7b. Presented at (for conference papers) Title, place and date of conference				
8. Author 1 Surname, initials Laycock D P	9(a) Author 2 Powell W J A	9(b) Authors 3,4...	10. Date 1968.5	pp. ref. 40
11. Contract Number	12. Period	13. Project	14. Other Reference	
15. Distribution statement Unlimited				
Descriptors (or keywords)				
continue on separate piece of paper				
<p>Abstract: Philips Usfa BV Stirling cycle split-mode cooling engines, sampled from development, pre-production and production batches, have been extensively tested at RSRE. Measurement of cryogenic performance at normal and high ambient temperatures in run-stop-run and continual-run regimes have been made. Representative detector encapsulations have been interfaced to provide typical combination heat loads. One pre-production unit has been installed and operated successfully in a Thermal Imaging Common Module (TICM) Class 2 Scanner. Detector microphony-noise measurements were made and compared with those recorded when the same array was cooled with high pressure pure gas liquified by a Hymatic self-regulating mini-cooler. Initial problems of contamination of the helium gas and of the thermal interface with the detector encapsulation have been highlighted and addressed by the manufacturers. A production standard unit has achieved over 5,000 hours <u>intermittent and continuous running during a period of 12 months.</u></p>				